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MINE HAULAGE

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MECHANICS.

(PART 1.)

MATTER AND ITS PROPERTIES.

1799. **Matter** is anything that occupies space. It is the substance of which all bodies are composed. Matter is composed of *molecules* and *atoms*.

1800. A **molecule** is the smallest portion of matter than can exist without changing its nature.

1801. An **atom** is an indivisible portion of matter.

Atoms unite to form molecules, and a collection of molecules form a mass or body.

A drop of water may be divided and subdivided, until each particle is so small that it can only be seen by the most powerful microscope, but each particle will still be water. Now, imagine the division to be carried on still further, until a limit is reached beyond which it is impossible to go without changing the nature of the particle. The particle of water is now so small that, if it be divided again, it will cease to be water, and will be something else; we call this particle a *molecule*.

If a molecule of water be divided, it will yield two atoms of hydrogen gas, and one of oxygen gas. If a molecule of sulphuric acid be divided, it will yield two atoms of hydrogen, one of sulphur, and four of oxygen.

It has been calculated that the diameter of a molecule is larger than $\frac{1}{115000000}$ of an inch, and smaller than $\frac{1}{50000000}$ of an inch.

1802. **Bodies** are composed of collections of molecules. Matter exists in three conditions or forms: *solid, liquid,* and *gaseous.*

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1803. A **solid body** is one whose molecules change their relative positions with great difficulty; as iron, wood, stone, etc.

1804. A **liquid body** is one whose molecules tend to change their relative positions easily.   Liquids readily adapt themselves to the vessel which contains them, and their upper surface always tends to become perfectly level. Water, mercury, molasses, etc., are liquids.

1805. A **gaseous body,** or gas, is one whose molecules tend to separate from one another; as air, oxygen, hydrogen, etc.

Gaseous bodies are sometimes called **aeriform** (air-like) **bodies.**   They are divided into two classes—the so-called "*permanent*" *gases* and *vapors.*

A **permanent gas** is one which remains a gas at ordinary temperatures and pressures.

A **vapor** is a body which, at ordinary temperatures, is a liquid or solid, but, when heat is applied, becomes a gas, as steam.

1806. One body may be in all three states; as, for example, mercury, which at ordinary temperatures is a liquid, becomes a solid (freezes) at 40° below zero, and a vapor (gas) at 600° above zero.   By means of great cold, all gases, even hydrogen, have been liquefied, and some solidified.

By means of heat, all solids have been liquefied, and a great many vaporized.   It is probable that, if we had the means of producing sufficiently great extremes of heat and cold, all solids might be converted into gases, and all gases into solids.

1807. Every portion of matter possesses certain qualities called *properties.*   Properties of matter are divided into two classes, *general* and *special.*

General properties of matter are those which are common to all bodies.   They are as follows: *Extension, impenetrability, weight, indestructibility, inertia, mobility,*

divisibility, porosity, compressibility, expansibility, and *elasticity.*

1808. Extension is the property of occupying space. Since all bodies must occupy space, it follows that extension is a general property.

By **impenetrability** we mean that no two bodies can occupy exactly the same space at the same time.

1809. Weight is the measure of the earth's attraction upon a body. All bodies have weight. In former times it was supposed that gases had no weight, since, if unconfined, they tend to move away from the earth, but, nevertheless, they will finally reach a point beyond which they can not go, being held in suspension by the earth's attraction. Weight is measured by comparison with a standard. The standard is a bar of platinum owned and kept by the Government; it weighs one pound.

1810. Inertia means that a body can not put itself in motion nor bring itself to rest. To do either, it must be acted upon by some force.

1811. Mobility means that a body can be changed in position by some force acting upon it.

1812. Divisibility is that property of matter which indicates that a body may be separated into parts.

1813. Porosity is that property of matter which indicates that there is space between the molecules of a body. Molecules of a body are supposed to be spherical, and, hence, there is space between them, as there would be between peaches in a basket. The molecules of water are larger than those of salt; so that when salt is dissolved in water, its molecules wedge themselves between the molecules of the water, and unless too much salt is added, the water will occupy no more space than it did before. This does not prove that water is penetrable, for the molecules of salt occupy the space that the molecules of water did not.

Water has been forced through iron by pressure, thus proving that iron is porous.

1814. Compressibility is that property of matter which indicates that the molecules of a body may be crowded nearer together, so as to occupy a smaller space.

1815. Expansibility is that property of matter which indicates that the molecules of a body may be forced apart, so as to occupy a greater space.

1816. Elasticity is that property of matter which indicates that if a body be distorted within certain limits, it will resume its original form when the distorting force is removed. Glass, ivory, and steel are very elastic.

1817. Indestructibility indicates that matter can never be destroyed. A body may undergo thousands of changes; be resolved into its molecules, and its molecules into atoms, which may unite with other atoms to form other molecules and bodies entirely different from the original body, but the same number of atoms remain. The whole number of atoms in the universe is exactly the same now as it was millions of years ago, and will always be the same. *Matter is indestructible.*

1818. Special properties are those which are not possessed by all bodies. Some of the most important are as follows : *Hardness, tenacity, brittleness, malleability,* and *ductility.*

1819. Hardness is that property of matter which indicates that some bodies may scratch other bodies. Fluids and gases do not possess hardness. The diamond is the hardest of all substances.

1820. Tenacity is that property of matter which indicates that some bodies resist a force tending to pull them apart. Steel is very tenacious.

1821. Brittleness is that property of matter which indicates that some bodies are easily broken; as glass, crockery, etc.

1822. Malleability is that property of matter which indicates that some bodies may be hammered or rolled into sheets. Gold is the most malleable of all substances.

1823. Ductility is that property of matter which in- . dicates that some bodies may be drawn into wire. Platinum is the most ductile of substances.

MOTION AND VELOCITY.

1824. Motion is the opposite of rest, and indicates a changing of position in relation to some object. If a large stone is rolled down hill, it is in motion in relation to the hill.

If a person is on a railway-train, and walks in the opposite direction from that in which the train is moving, and with the same speed, he will be in motion as regards the train, but at rest with respect to the earth, since, until he gets to the end of the train, he will be directly over the spot at which he was when he started to walk.

1825. The **path** of a body in motion is the line described by its *central point.* No matter how irregular the shape of the body may be, nor how many turns and twists it may make, the line which indicates the direction of the center of the body for every instant that it was in motion is the path of the body.

1826. Velocity is rate of motion. It is measured by a unit of space passed over in a unit of time. When equal spaces are passed over in equal times, the velocity is said to be **uniform.** In all other cases, it is **variable.**
- If the fly-wheel of an engine keeps up a constant speed of a certain number of revolutions per minute, the velocity of any point is uniform. A railway-train having a constant speed of 40 miles per hour moves 40 miles every hour, or $\frac{40}{60} = \frac{2}{3}$ of a mile every minute, and since equal spaces are passed over in equal times, the velocity is uniform.

1827. To find the uniform velocity which a body must have to pass over a certain distance or space in a given time :

Rule.—*Divide the distance by the time.*

Let s = distance traveled by moving body;

v = uniform velocity of body;

t = the time.

Then, $$v = \frac{s}{t}. \quad \textbf{(90.)}$$

EXAMPLE.—The piston of a steam-engine travels 3,000 feet in 5 minutes; what is its velocity in feet per minute?

SOLUTION.—Here 3,000 feet is the distance, and 5 minutes is the time. Applying formula **90**,

$$v = \frac{s}{t} = \frac{3,000}{5} = 600 \text{ feet per minute.} \quad \text{Ans.}$$

CAUTION.—Before applying the above or any of the succeeding rules, care must be taken to reduce the values given to the denominations required in the answer. Thus, had the velocity been required to have been in feet per second instead of feet per minute in the above example, the 5 minutes should first be reduced to seconds before dividing. The operation would then have been 5 min. $= 5 \times 60 = 300$ sec. Then, according to the formula,

$$v = 3,000 \div 300 = 10 \text{ ft. per sec.} \quad \text{Ans.}$$

Had the velocity been required in inches per second, it would have been necessary to reduce the 3,000 feet to inches and the 5 minutes to seconds before dividing. Thus, 3,000 ft. $\times$ 12 = 36,000 in. 5 min. $\times$ 60 = 300 sec. Now, applying the formula,

$$v = \frac{36,000}{300} = 120 \text{ in. per sec.} \quad \text{Ans.}$$

EXAMPLE.—A railroad-train travels 50 miles in 1½ hours; what is its average velocity in feet per second?

SOLUTION.—Reducing the miles to feet and the hours to seconds, 50 miles $\times$ 5,280 = 264,000 ft. 1½ hours $\times$ 60 $\times$ 60 = 5,400 sec. Applying formula **90**,

$$v = \frac{264,000}{5,400} = 48\tfrac{8}{9} \text{ ft. per sec.} \quad \text{Ans.}$$

1828. To find the distance which a body would travel in a given time with a given velocity:

Rule.—*Multiply the velocity by the time,*

or $$s = v\,t. \quad \textbf{(91.)}$$

EXAMPLE.—The velocity of sound in still air is 1,092 feet per second; how many miles will it travel in 16 seconds ?

SOLUTION.—Reducing the 1,092 ft. to miles, the velocity is

$$\frac{1,092}{5,280} \text{ mile per second.}$$

Applying formula **91**,

$$s = v\,t = \frac{1,092}{5,280} \times 16 = 3.31 \text{ miles, nearly.} \quad \textbf{Ans.}$$

EXAMPLE.—The piston speed of an engine is 11 ft. per sec.; how many miles does the piston travel in 1 hour and 15 minutes ?

SOLUTION.— 1 hour and 15 minutes reduced to seconds = 4,500 seconds = the time. 11 feet reduced to miles = $\dfrac{11}{5,280}$ mile = velocity in miles per second. Applying the formula,

$$s = \frac{11}{5,280} \times 4,500 = 9.375 \text{ miles.} \quad \textbf{Ans.}$$

1829. To find the time it will take a body to move through a given distance with a given uniform velocity:

Rule.—*Divide the distance, or space passed over, by the velocity.*

$$t = \frac{s}{v}. \qquad \textbf{(92.)}$$

EXAMPLE.—Suppose that the radius of the crank of a steam-engine is 15 inches and that the shaft makes 120 revolutions per minute, how long will it take the crank-pin to travel 18,849.6 feet ?

SOLUTION.—Since the radius, or distance from the center of the shaft to the center of the crank-pin, is 15 in., the diameter of the circle it moves in is 15 in. × 2 = 30 in. = 2.5 ft. The circumference of this circle is 2.5 × 3.1416 = 7.854 ft. 7.854 × 120 = 942.48 ft., distance that the crank-pin travels in one minute = velocity in feet per minute. Applying the formula,

$$t = \frac{s}{v} = \frac{18,849.6}{942.48} = 20 \text{ min.} \quad \text{Ans.}$$

EXAMPLE.—A point on the rim of an engine fly-wheel travels at the rate of 150 feet per second; how long will it take to travel 45,000 feet ?

SOLUTION.—Using formula **92**,

$$t = \frac{45,000}{150} = 300 \text{ sec.} = 5 \text{ min.} \quad \text{Ans.}$$

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EXAMPLES FOR PRACTICE.

1. A locomotive has drivers 80 inches in diameter. If they make 208 revolutions per minute, what is the velocity of the train in (*a*) feet per second? (*b*) miles per hour?   Ans. $\begin{cases} (a)\ 102.277\ \text{ft. per sec.} \\ (b)\ 69.734\ \text{mi. per hr.} \end{cases}$

2. Assuming the velocity of steam as it enters the cylinder to be 900 feet per second, how far could it travel, if unobstructed, during the time the fly-wheel of an engine revolved 7 times, if the number of revolutions per minute were 120?   Ans. 3,150 ft.

3. The average speed of the piston of an engine is 528 feet per minute; how long will it take the piston to travel 4 miles?
Ans. 40 min.

4. A speed of 40 miles per hour equals how many feet per second?
Ans. 58⅔ ft.

5. The earth turns around once in 24 hours. If the diameter be taken as 8,000 miles, what is the velocity of a point on the earth in miles per minute?   Ans. 17.45⅓ mi. per min.

6. The stroke of an engine is 23 inches. If the engine makes 11,400 strokes per hour, (*a*) what is its speed in feet per minute? (*b*) How far will this piston travel in 11 minutes?   Ans. $\begin{cases} (a)\ 443\tfrac{1}{3}\ \text{ft. per min.} \\ (b)\ 4,876\ \text{ft. 8 in.} \end{cases}$

FORCE.

NEWTON'S LAWS OF MOTION.

1830. A **force** is that which produces, or tends to produce or destroy, motion. Forces are called by various names, according to the effects which they produce upon a body, as *attraction, repulsion, cohesion, adhesion, accelerating* force, *retarding* force, *resisting* force, etc., but all are equivalent to a push or pull, according to the direction in which they act upon a body.

1831. That the effect of a force upon a body may be compared with another force, it is necessary that three conditions be fulfilled in regard to both bodies. They are as follows:

1. *The point of application, or point at which the force acts upon the body, must be known.*

2. *The direction of the force, or, what is the same thing, the straight line along which the force tends to move the point of application, must be known.*

3. *The magnitude or value of the force, when compared with a given standard, must be known.*

The unit of magnitude of forces will always be taken as one pound, and all forces will be spoken of as a certain number of pounds.

1832. In practice, force is always regarded as a pressure; that is, a force may always be replaced by an equivalent weight. Thus, a force of 20 lb. acting upon a body is regarded as a pressure of 20 lb. produced by a weight of 20 lb. The tendency of a force is always to produce motion in the direction in which it acts. The resistance may be too great to cause motion, but it *always tends* to produce it.

1833. The fundamental principles of the relations between force and motion were first stated by Sir Isaac Newton. They are called "Newton's Three Laws of Motion," and are as follows:

I. *All bodies continue in a state of rest, or of uniform motion in a straight line, unless acted upon by some external force that compels a change.*

II. *Every motion or change of motion is proportional to the acting force, and takes place in the direction of the straight line along which the force acts.*

III. *To every action there is always opposed an equal and contrary reaction.*

1834. In the *first law of motion* it is stated that a body once set in motion by any force, no matter how small, will move forever in a straight line, and always with the same velocity, unless acted upon by some other force which compels a change. It is not possible to actually verify this law, on account of the earth's attraction for all bodies, but from astronomical observations, we are certain that the law is true. This law is often called *the law of inertia.*

1835. The word **inertia** is so abused that a full understanding of its meaning is necessary. Inertia is not a force, although it is often so called. If a force acts upon a body and puts it in motion, the effect of the force is stored in the body, and a second body, in stopping the first, will receive a blow equal in every respect to the original force, assuming that there has been no resistance of any kind to the motion of the first body.

It is dangerous for a person to jump from a fast-moving train, for the reason that, since his body has the same velocity as the train, it has the same force stored in it that would cause a body of the same weight to take the same velocity as the train, and the effect of a sudden stoppage is the same as the effect of a blow necessary to give the person that velocity.

By "bracing" himself and jumping in the same direction that the train is moving, and running, he brings himself gradually to rest; and thus reduces the danger. If a body is at rest, it must be acted upon by a force in order to be.put in motion, and no matter how great the force may be, it can not be *instantly* put in motion.

The resistance thus offered to being put in motion is commonly, but erroneously, called the *resistance of inertia.* It should be called the *resistance due to inertia.*

1836. From the *second law*, it is seen that, if two or more forces act upon a body, their final effect upon the body will be in proportion to their magnitude and to the directions in which they act. Thus, if the wind is blowing due west, with a velocity of 50 miles per hour, and a ball is thrown due north with the same velocity, or 50 miles per hour, the wind will carry the ball just as far west as the force of the throw carried it north, and the combined effect will be to cause it to move northwest. The amount of departure from due north will be proportional to the force of the wind, and independent of the velocity due to the force of the throw.

1837. In Fig. 587, a ball e is supported in a cup, the bottom of which is attached to the lever o in such a manner that a movement of o will swing the bottom horizontally and allow the ball to drop. Another ball b rests in a horizontal groove that is provided with a slit in the bottom. A swinging arm is actuated by the spring d in such a manner that, when drawn back as shown and then released, it will strike the lever o and the ball b at the same time. This gives b an impulse in a horizontal direction and swings o so as to allow e to fall.
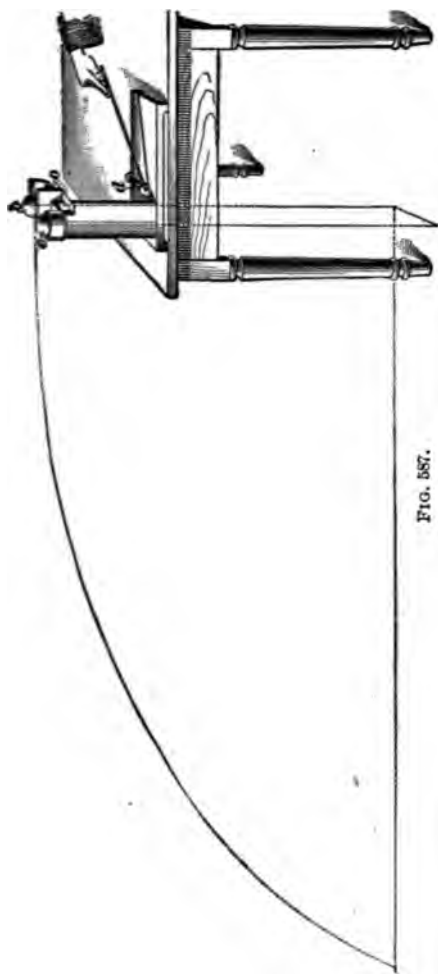
On trying the experiment, it is found that b follows a path shown by the curved dotted line, and reaches the floor at the same instant as e, which drops vertically. This shows that the force which gave the first ball its horizontal movement had no effect on the vertical force which compelled both balls to fall to the floor, the vertical force producing the same effect as if the horizontal force had not acted. The second law may also be stated as follows: *A force has the same effect in producing motion, whether it acts upon a body at*

FIG. 587.

rest or in motion, and whether it acts alone or with other forces.

1838. The *third law* states that action and reaction are equal and opposite. A man can not lift himself by his boot-straps, for the reason that he presses downwards with the same force that he pulls upwards; the downward reaction equals the upward action, and is opposite to it.

In springing from a boat, we must exercise caution, or the reaction will drive the boat from the shore. When we jump from the ground, we tend to push the earth from us, while the earth reacts and pushes us from it.

EXAMPLE.—Two men pull on a rope in opposite directions, each exerting a force of 100 pounds; what is the force which the rope resists?

SOLUTION.—Imagine the rope to be fastened to a tree, and one man to pull with a force of 100 pounds. The rope evidently resists 100 pounds. According to Newton's third law, the reaction of the tree is also 100 pounds. Now, suppose the rope to be slackened, but that one end is still fastened to the tree, and the second man to take hold of the rope near the tree, and pull with a force of 100 pounds, the first man pulling as before. The resistance of the rope is 100 pounds, as before, since the second man merely takes the place of the tree. *He is obliged to exert a force of 100 pounds to keep the rope from slipping through his fingers.* If the rope be passed around the tree and each man pulls an end with a force of 100 pounds in the same and parallel directions, the stress in the rope is 100 pounds, as before, but the tree must resist the pull of both men, or 200 pounds.

1839. A **force** may be represented by a line; thus, in Fig. 588, let *A* be the *point of application* of the force; let

A———▶———*B*

the length of the line *A B* represent its *magnitude*, and let the arrow-head indicate

FIG. 588.

the *direction* in which the force acts; then the line *A B* fulfils the three conditions (see Art. **1831**), and the force is fully represented.
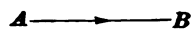
CENTER OF GRAVITY.

1840. The **center of gravity** of a body is that point at which the body may be balanced, or it is the point at which the whole weight of a body may be considered as concentrated.

1841. In a moving body, the line described by its center of gravity is always taken as the path of the body. In finding the distance that a body has moved, the distance that the center of gravity has moved is taken.

The definition of the center of gravity of a body may be applied to a system of bodies, if they are considered as being connected at their centers of gravity.

1842. If w and W, Fig. 589, be two bodies of known weights, their center of gravity will be at C. The point C may be readily deter-
mined, as follows:

Rule.—*The dis-
tance of the common
center of gravity from
the center of gravity of
the large weight is
equal to the weight of
the smaller body mul-*
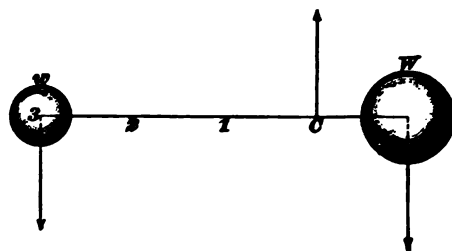
FIG. 589.

tiplied by the distance between the centers of gravity of the two bodies, and this product divided by the sum of the weights of the two bodies.

Let w = weight of smaller body;

        W = weight of larger body;

        l = distance between centers of gravity of the two bodies;

        l_1 = distance from the center of gravity of the two to the center of gravity of the larger body.

Then, $$l_1 = \frac{w\,l}{W + w}.$$ **(93.)**

EXAMPLE.—In Fig. 589, w = 10 pounds, W = 30 pounds, and the distance between their centers of gravity is 36 inches; where is the center of gravity of both bodies situated ?

SOLUTION.—Applying formula **93**,

$$l_1 = \frac{10 \times 36}{30 + 10} = 9 \text{ in.} =$$

distance of center of gravity from center of large weight. Ans.

1843. It is now very easy to extend this principle, to find the center of gravity of any number of bodies when their weights and the distances apart of their centers of gravity are known, by the following rule:

Rule.—*Find the center of gravity of two of the bodies, as W_1 and W_4, in Fig. 590, at C_1. Assume that the weight of both bodies is concentrated at C_1, and find the center of gravity of this combined weight C_1, and the weight of W_2 to be at C_2; then, find that the center of gravity of the combined weights of W_1, W_4, and W_2 (concentrated at C_2) and W_3 to be at C, and C will be the center of gravity of the four bodies.*
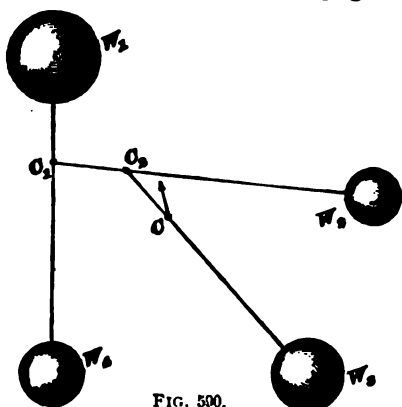
FIG. 590.

1844. To find the center of gravity of **any parallelogram:**

Rule.—*Draw the two diagonals, Fig. 591, and their point of intersection C will be the center of gravity.*
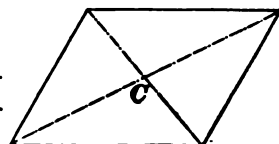
FIG. 591.

1845. To find the center of gravity of a **triangle,** as $A B C$, Fig. 592:

Rule.—*From any vertex, as A, draw a line to the middle point D of the opposite side $B C$. From one of the other vertexes, as C, draw a line to F, the middle point of the opposite side $A B$; the point of intersection O of these two lines is the center of gravity.*
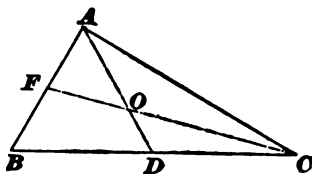
FIG. 592.

It is also true that the distance $D O = \frac{1}{3} D A$, and that

$F O = \frac{1}{3} F C$, and the center of gravity could have been found by drawing from any vertex a line to the middle point of the opposite side, and measuring back from that side $\frac{1}{3}$ of the length of the line.

1846. The center of gravity of **any regular plane figure** is the same as the center of the inscribed or circumscribed circle.

1847. To find the center of gravity of **any irregular plane figure** but of uniform thickness throughout, divide one of the parallel surfaces into triangles, parallelograms,
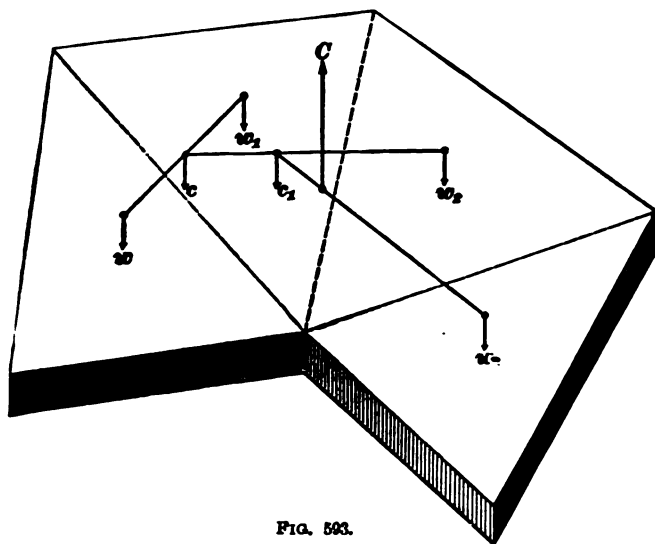
Fig. 593.

circles, ellipses, etc., according to the shape of the figure; find the area and center of gravity of each part separately, and combine the centers of gravity thus found as in the case of more than two bodies whose weights were known by the rule of Art. **1843,** except that the area of each part is used instead of their weights. See Fig. 593.

EXAMPLE.—Suppose that the two balls shown in Fig. 589 are 5 inches and 10 inches in diameter, and weigh 10 pounds and 80 pounds, respectively. If the distance between their centers is 40 inches, and

they are connected by a steel rod 1 inch in diameter, where is the center of gravity, taking the weight of a cubic inch of steel as .283 pound?

SOLUTION.—The length of the rod $= 40 - \frac{5}{2} - \frac{12}{2} = 32\frac{1}{2}$ in. Its volume is $1^2 \times .7854 \times 32\frac{1}{2} = 25.53$ cu. in. $25.53 \times .283 = 7.22$ lb. The rod being straight, its center of gravity is in the middle at a distance of $\frac{32.5}{2} + \frac{5}{2} = 18\frac{3}{4}$ in. from the center of the smaller weight and $\frac{32.5}{2} + \frac{12}{2} = 21\frac{1}{4}$ in. from the center of the larger weight. Now, assuming the weight of the rod to be concentrated at its center of gravity, we have three weights of 10, 7.22, and 80 lb., all in a straight line, and the distances between them given, to find the center of gravity, or balancing-point, of the combination. We will first find the center of gravity of the two smaller weights by formula **93.**

$$l_1 = \frac{7.22 \times 18\frac{3}{4}}{10 + 7.22} = 7.86 \text{ in.} =$$

distance from the center of the 10-lb. weight. Considering both of the smaller weights to be concentrated at this point, we find the center of gravity of this combined weight and the large weight by the same formula:

$$40 - 7.86 = 32.14 \text{ in.} =$$

distance between the center of gravity of the two small weights and the center of gravity of the 80-lb. weight. Applying formula **93,**

$$l_1 = \frac{17.22 \times 32.14}{80 + 17.22} = 5.693 \text{ in.} =$$

distance from the center of the 80-lb. weight. Ans.

1848. Center of Gravity of a Solid.—In a body free to move, the center of gravity will lie in a vertical plumb-line
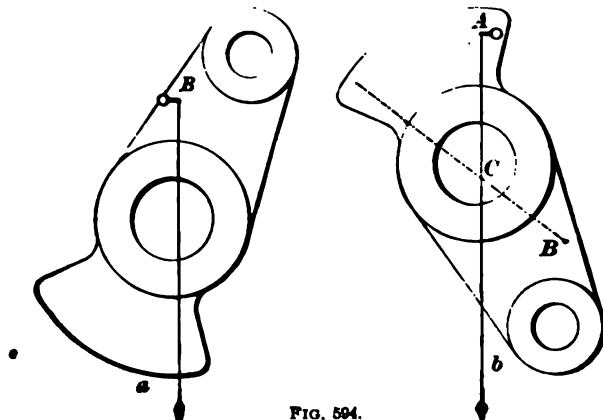
FIG. 594.

drawn through the point of support. Therefore, to find the position of the center of gravity of an irregular solid, as the crank, Fig. 594, suspend it at some point, as *B*, so that it will move freely. Drop a plumb-line from the point of suspension, and mark its direction. Suspend the body at another point, as *A*, and repeat the process. The intersection *C* of the two lines will be directly over the center of gravity.

Since the center of gravity depends wholly upon the shape and weight of a body, it may be without the body; for example, the center of gravity of a circular ring is the same as the center of the circumference of the ring.

EXAMPLES FOR PRACTICE.

1. A spherical shell has a wrought-iron handle attached to it. The shell is 10 inches in diameter, and weighs 20 pounds. The handle is 1¼ inches in diameter, and the distance from the center of the shell to the end of the handle is 4 feet. Where is the center of gravity? Take the weight of a cubic inch of wrought iron as .278 pound.

Ans. 13.612 in. from center of shell.

2. The distance between the centers of two bodies is 51 inches. The weights of the bodies being 20 and 73 pounds, where is the center of gravity? Ans. 10.968 in. from the center of large weight.

3. A hollow engine piston weighs 275 pounds, and is 3¼ inches thick. Assuming the piston-rod to be straight throughout its entire length, and to weigh 140 pounds, at what point will the piston and rod balance, if the length of the rod is 73 inches from the face of the piston? Consider the weight of the piston to be concentrated at its center.

Ans. 11.15 in., nearly, from face of piston.

4. Weights of 5, 9, and 12 pounds lie in one straight line, in the order named. Distance from the 5-pound weight to the 9-pound weight is 22 inches, and from the 9-pound weight to the 12-pound weight is 18 inches. Where is the center of gravity?

Ans. 13.923 in. from 12-pound weight.

SIMPLE MACHINES.

THE LEVER AND WHEEL AND AXLE.

1849. A **lever** is a bar capable of being turned about a pivot, or point, as in Figs. 595 to 597.

The object *W* to be lifted is called the **weight**; the force used *P* is called the **power**; and the point or pivot *F* is called the **fulcrum.**

That part of the lever between the weight and the ful-
crum, or Fb, is called the **weight arm,** and the part be-

FIG. 595.　　　　　FIG. 596.　　　　　FIG. 597.
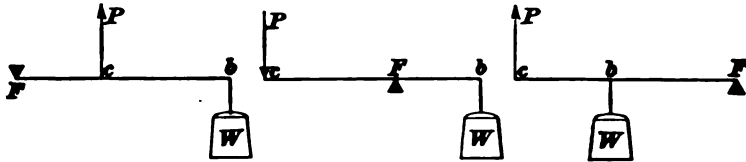
tween the power and the fulcrum, or Fc, is called the
power arm.

1850. In order that the lever shall be in equilibrium
(balance), *the power multiplied by the power arm must equal
the weight multiplied by the weight arm;* that is, $P \times Fc$
must equal $W \times Fb$.

If F be taken as the center of a circle, and arcs be de-
scribed through b and c, it will be seen that, if the weight
arm is moved through a certain angle, the power arm will
move through the same angle. Since, in the same or equal
angles, the lengths of the arcs are proportional to the radii
with which they were described, it is seen that the power
arm is proportional to the distance through which the power
moves, and the weight arm is proportional to the distance
through which the weight moves.

Hence, instead of writing $P \times Fc = W \times Fb$, we might
have written it $P \times$ distance through which P moves $= W \times$
distance through which W moves. This is the general law
of all machines, and can be applied to any mechanism, from
the simple lever up to the most complicated arrangement.
Stated in the form of a rule, it is as follows:

Rule.—*The power multiplied by the distance through which
it moves equals the weight multiplied by the distance through
which it moves.*

In the above rule, it will be noticed that there are four
requirements necessary for a complete knowledge of the
lever, viz., the power (or force), the weight, the power arm
(or distance through which the power moves), and the weight

arm (or distance through which the weight moves). If any three are given, the fourth may be found by letting x represent the requirement which is to be found, and multiplying the power by the power arm and the weight by the weight arm; then, dividing the product of the two known numbers by the number by which x is multiplied, the result will be the requirement which was to be found.

EXAMPLE.—If the weight arm of a lever is 6 inches long, and the power arm is 4 feet long, how great a weight can be raised by a force of 20 pounds at the end of the power arm ?

SOLUTION.—In this example, the weight is unknown; hence, representing it by x, we have, after reducing the 4 ft. to inches, $20 \times 48 = 960 =$ power multiplied by the power arm, and $x \times 6 =$ weight multiplied by the weight arm. Dividing the 960 by 6, the result is 160 lb., the weight. Ans.

If the distance through which the power or weight moved had been given instead of the power arm or weight arm, and it were required to find the power or weight, the process would have been exactly the same, using the given distance instead of the power arm or weight arm.

EXAMPLE.—If, in the above example, the weight had moved $2\frac{1}{2}$ inches, how far would the power have moved ?

SOLUTION.—In this example, the distance through which the power moves is required. Let x represent the distance. Then, $20 \times x =$ distance multiplied by power, and $2\frac{1}{2} \times 160 = 400 =$ distance multiplied by the weight. Hence, $x = \frac{400}{20} = 20$ in. $=$ distance through which the power arm moves. Ans.

The ratio between the weights and the power is $160 \div 20 = 8$. The ratio between the distance through which the weight moves and the distance through which the power moves is $2\frac{1}{2} \div 20 = \frac{1}{8}$. This shows that while a force of 1 pound can raise a weight of 8 pounds, the 1-pound weight must move through 8 times the distance that the 8-pound weight does. It will also be noticed that the ratio of the lengths of the two arms of the lever is also 8, since $48 \div 6 = 8$.

1851. The law which governs the straight lever also governs the bent lever, but care must be taken to determine

the true lengths of the lever arms, which are in every
case *the perpendicular distances from the fulcrum to the line
of direction of the weight or power.*

Thus, in Figs. 598 to 601, Fc in each case represents the
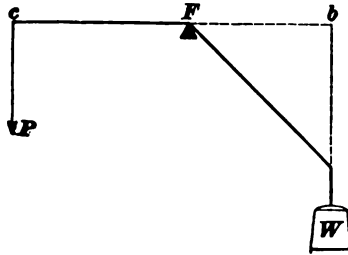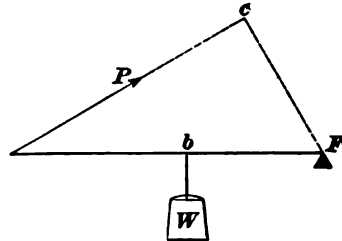
FIG. 598.   FIG. 599.
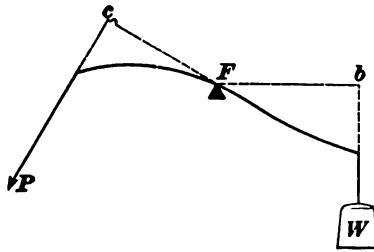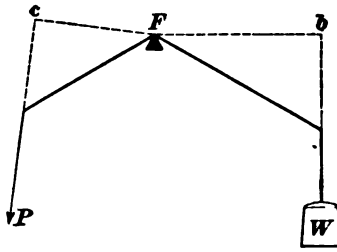
FIG. 600.   FIG. 601.

power arm, and Fb the weight arm.   The following formula
applies to any lever, straight or bent:

Let P = power;

    W = weight;

    a = perpendicular distance of line of direction of
power from fulcrum = power arm;

    b = perpendicular distance of line of direction of
weight from fulcrum = weight arm.

Then, $Pa = Wb.$ (94.)

1852. **A compound lever** is a series of single levers
arranged in such a manner that when a power is applied to
the first, it is communicated to the second, and from this to
the third, and so on.

Fig. 602 shows a compound lever. It will be seen that, when a power is applied to the first lever at P, it will be communicated to the second lever at P, from this to the third lever at P, and thus raise the weight W.

The weight which the power of the first lever could raise acts as the power of the second, and the weight which this could raise through the second lever acts as the power of the third lever, and so on, no matter how many single levers make up the compound lever.

In this case, as in every-other, the power multiplied by the distance through which it moves equals the weight multiplied by the distance through which it moves.

Hence, if we move the P end of the lever, say 4 inches,
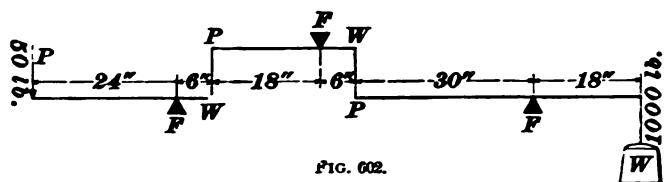
FIG. 602.

and the W end moves $\frac{1}{5}$ of an inch, we know that the ratio between P and W is the same as the ratio between 4 and $\frac{1}{5}$, that is, 1 to 20, and, hence, that 10 pounds at P would balance 200 pounds at W, without measuring the lengths of the different lever arms. If the lengths of the lever arms are known, the ratio between P and W may be readily obtained from the following rule:

Rule.—*The continued product of the power and each power arm equals the continued product of the weight and each weight arm.*

Let $a_1, a_2, a_3 \ldots$ = power arms of compound lever;

$\quad\quad b_1, b_2, b_3 \ldots$ = weight arms of compound lever.

Then,

$$P \times a_1 \times a_2 \times a_3 \times \ldots = W \times b_1 \times b_2 \times b_3 \times \ldots \quad \text{(95.)}$$

EXAMPLE.—If, in Fig. 602, $PF = 24$ inches, 18 inches, and 30 inches, respectively, and $WF = 6$ inches, 6 inches, and 18 inches, respectively, how great a force at P would it require to raise 1,000 pounds at W? What is the ratio between W and P?

SOLUTION.—Let x represent the power; then, according to formula **95**, $x \times 24 \times 18 \times 30 = 12,960\,x.$ $1,000 \times 6 \times 6 \times 18 = 648,000.$

$$x = \frac{648,000}{12,960} = 50 \text{ lb. Ans.}$$

$1,000 \div 50 = 20 = $ ratio of W to $P.$ Ans.

1853.   The **wheel and axle** consists of *two cylinders of different diameters, rigidly connected,* so that they turn together about a common axis, as in Fig. 603.   Then, as
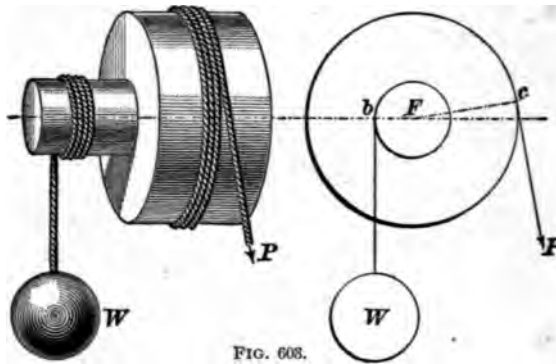
FIG. 603.

before, $P \times$ distance through which it moves $= W \times$ distance through which it moves; and, since these distances are proportional to the radii of the power cylinder and weight cylinder, $P \times F\,c = W \times F\,b.$

It is not necessary that an entire wheel be used; an arm, projection, radius, or anything which the power causes to revolve in a circle may be considered as the wheel.   Consequently, if it is desired to hoist a weight with a windlass, Fig. 604, the power is applied to the handle of the crank, and the distance between the center line of the crank handle and the axis of the drum corresponds to the radius of the wheel.
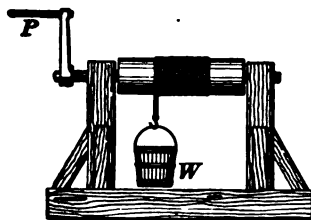
FIG. 604.

EXAMPLE.—If the distance between the center line of the handle and the axis of the drum, in Fig. 604, is 18 inches, and the diameter of the drum

is 6 inches, what force will be required at P to raise a load of 300 pounds ?

SOLUTION.—$P \times (18 \times 2) = 300 \times 6$, or $P = 50$.   Ans.

EXAMPLES FOR PRACTICE.

1. The lever of a safety-valve is of the form shown in Fig. 595, where the force is applied at a point between the fulcrum and the weight lifted. If the distance from the fulcrum to the valve is $5\frac{1}{4}$ inches, and from the fulcrum to the weight is 42 inches, what total force is necessary to raise the valve, the weight being 78 pounds and the weight of valve and lever being neglected?   Ans. 595.64 lb.

2. If, in Fig. 602, $P\,F = 10, 12, 14$, and 16 inches, respectively, and $W\,F = 2, 3, 4$, and 5 inches, respectively, (a) how great a weight can a force of 20 pounds raise? (b) What is the ratio between W and P? (c) If P moves 4 inches, how far will W move?   Ans. $\begin{cases} (a) \ 4,480 \text{ lb.} \\ (b) \ 224. \\ (c) \ \frac{1}{56} \text{ in.} \end{cases}$

3. A windlass is used to hoist a weight. If the diameter of the drum on which the rope winds is 4 inches, and the distance from the center of the handle to the axis of the drum is 14 inches, how great a weight can a force of 32 pounds applied to the handle raise?
          Ans. 224 lb.

PULLEYS AND GEARS.

FIXED AND MOVABLE PULLEYS.

1854. A **pulley** is a wheel turning on an axle, over which a cord, chain, or band is passed in order to transmit the force through the cord, chain, or band.

Pulleys are often used for hoisting or raising loads, in which case the frame which supports the axle of the pulley is called the **block.**

FIG. 605.

1855. A **fixed pulley** is one whose block is not movable (see Fig. 605). In this case, if the weight W be lifted by pulling down P, the other end of the cord W will evidently move the same distance upwards that P moves downwards; hence, P must equal W.
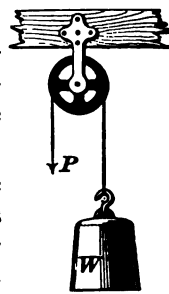
1856. A **movable pulley** is one whose block is movable, as in Fig. 606. One end of the cord is fastened to the

beam, and the weight is suspended from the pulley, the other end of the cord being drawn up by the application of a force P. A little consideration will show that if P moves through a certain distance, say 1 foot, W will move through *half* that distance, or 6 inches; hence, a pull of 1 pound at P will lift 2 pounds at W.
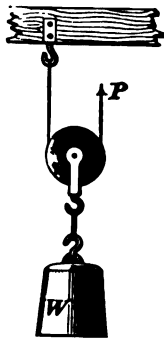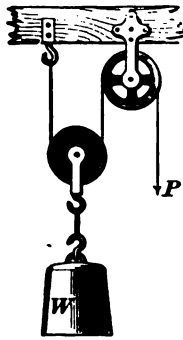
The same would also be true if the free end of the cord were passed over a *fixed pulley*, as in Fig. 607, in which case the fixed pulley merely changes the direction in which P acts, so that a weight of 1 pound hung on the free end of the cord will balance 2 pounds hung from the *movable pulley*.

FIG. 606.   FIG. 607.

1857. **A combination of pulleys,** as shown in Fig. 608, is sometimes used. In this case, there are three movable and three fixed pulleys, and the amount of movement of W, owing to a certain movement of P, is readily found.

It will be noticed that there are *six parts* of the rope, not counting the free end; hence, if the movable block be lifted 1 foot, P remaining in the same position, there will be 1 foot of slack in each of the six parts of the rope, or *six feet* in all. Therefore, P would have to move 6 feet in order to take up this slack, or P moves 6 times as far as W. Hence, 1 pound at P will support 6 pounds at W, since the *power multiplied by the distance through which it moves equals the weight multiplied by the distance through which it moves.* It will also be noticed that there are three movable pulleys, and that $3 \times 2 = 6$.
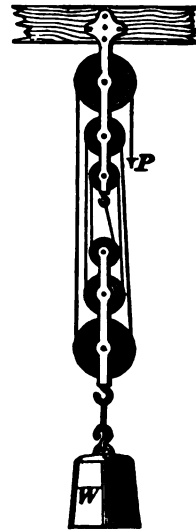
FIG. 608.

1858. Law of Combination of Pulleys.—*In any combination of pulleys where one continuous rope is used, a load on the free end will balance a weight on the movable block as many times as great as itself as there are parts of the rope supporting the load—not counting the free end.*

The above law is good, whether the pulleys are side by side, as in the ordinary block and tackle, or whether they are arranged as in the figure.

EXAMPLE.—In a block and tackle having five movable pulleys, how great a force must be applied to the free end of the rope to raise 1,250 pounds ?

SOLUTION.—Since there are five movable pulleys, there must be 10 parts of the rope to support them. Hence, according to the above law, a force applied to the free end will support a load 10 times as great as itself, or the force $= \dfrac{1,250}{10} = 125$ lb. Ans.

PULLEYS FOR TRANSMISSION OF POWER.

1859. Pulleys for the transmission of power by belts may be divided into two principal classes: (1) The solid

FIG. 609.   FIG. 610.

pulley shown in Fig. 609, in which the hub, arms, and rim are one entire casting. (2) The split pulley shown in Fig. 610, which is cast in halves.

This last style of pulley is more readily placed upon and removed from the shaft than the solid pulley. Pulleys are generally cast in halves or parts when they are more than 6 feet in diameter; this is done on account of shrinkage strain in large pulley castings, which renders them liable to crack as a result of unequal cooling of the metal.

1860. Crowning.—In Fig. 611 is shown a section of the rim of a pulley that has crowning, or, in other words, whose diameter is larger at the center of the face than at its edges. This is done to prevent the belt from running off the pulley. The amount of crowning given to pulleys varies from $\frac{3}{16}$ to $\frac{1}{2}$ an inch per foot of width of the pulley face.
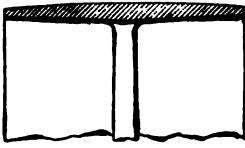
Fig. 611.

1861. Balanced Pulleys.—All pulleys which rotate at high speeds should be balanced. If they are not, the centrifugal force which is generated by the pulley's rotation is greater on one side than on the other, and it will cause the pulley shaft to vibrate and shake. Pulleys should run true, so that the strain or tension of the belt is equal at all parts of the revolution, thus making the transmitting power equal. The smoother the surface of a pulley, the greater is its driving power.

The transmitting power of a pulley can be increased by covering the face of the pulley with a leather or rubber band; this increases the driving power about one-quarter.

1862. The pulley that imparts motion to the belt is called the **driver;** that which receives the motion is called the **driven.**

The revolutions of any two pulleys over which a belt is run vary in an inverse proportion to their diameters; consequently, if a pulley of 20 inches in diameter is driven by one of 10 inches in diameter, the 20-inch pulley will make one revolution while the 10-inch pulley makes **two revolutions,** or they are in the ratio of 2 to 1.

1863. To find the diameter of the driving pulley, when the diameter of the driven pulley and the number of revolutions per minute of each are given:

Rule.—*The diameter of the driving pulley equals the product of the diameter and number of revolutions of the driven pulley, divided by the number of revolutions of the driving pulley.*

Let $D =$ diameter of the driver;
    $d =$ diameter of the driven;
    $N =$ number of revolutions of the driver;
    $n =$ number of revolutions of the driven.

NOTE.—The words revolutions per minute are frequently abbreviated to R. P. M.

Then, $D = \dfrac{d\,n}{N}.$ **(96.)**

EXAMPLE.—The driving pulley makes 100 revolutions per minute; the driven pulley makes 75 revolutions per minute, and is 18 inches in diameter; what is the diameter of the driving pulley?

SOLUTION.—Substituting in formula **96**, we have

$$D = \frac{18 \times 75}{100} = 13\tfrac{1}{2} \text{ in.} \quad \text{Ans.}$$

1864. The diameter and number of revolutions per minute of the driving pulley being given, to find the diameter of the driven pulley, which must make a given number of revolutions per minute:

Rule.—*The diameter of the driven pulley equals the product of the diameter and number of revolutions of the driving pulley, divided by the number of revolutions of the driven pulley.*

That is, $d = \dfrac{D\,N}{n}.$ **(97.)**

EXAMPLE.—The diameter of the driver is 13½ inches, and makes 100 revolutions per minute; what must be the diameter of the driven to make 75 revolutions per minute?

SOLUTION.—Substituting in formula **97**, we have

$$d = \frac{13\tfrac{1}{2} \times 100}{75} = 18 \text{ in.} \quad \text{Ans.}$$

1865. To find the number of revolutions per minute of the driven pulley, its diameter and the diameter and number of revolutions per minute of the driving pulley being given:

Rule.—*The number of revolutions of the driven pulley is equal to the product of the diameter and number of revolutions of the driver, divided by the diameter of the driven pulley.*

That is, $n = \dfrac{DN}{d}.$ **(98.)**

EXAMPLE.—The driver is 13¼ inches in diameter, and makes 100 revolutions per minute; how many revolutions will the driven make in one minute, if it is 18 inches in diameter ?

SOLUTION.—Substituting in formula **98,** we have

$$n = \frac{13\frac{1}{4} \times 100}{18} = 75 \text{ R. P. M.} \text{Ans.}$$

1866. To find the number of revolutions per minute of the driving pulley, its diameter and the diameter and number of revolutions per minute of the driven pulley being given:

Rule.—*The number of revolutions of the driving pulley is equal to the product of the diameter and number of revolutions of the driven pulley, divided by the diameter of the driving pulley.*

That is, $N = \dfrac{dn}{D}.$ **(99.)**

EXAMPLE.—The driven pulley is 18 inches in diameter, and makes 75 revolutions per minute; how many revolutions will the driver make in one minute, if it is 13¼ inches in diameter ?

SOLUTION.—Substituting in formula **99,** we have

$$N = \frac{18 \times 75}{13\frac{1}{4}} = 100 \text{ R. P. M.} \text{Ans.}$$

WHEELWORK.

1867. Wheelwork. — A combination of wheels and axles, as in Fig. 612, is called a **train.** The wheel in a train, to which motion is imparted from a wheel on another shaft by such means as a belt or gearing, is called the **driven wheel** or **follower;** the wheel which imparts the motion is called the **driver.**

It will be seen that the wheel and axle bears the same

relation to the train that the simple lever does to the com
pound lever. Letting D_1, D_2, D_3, etc., represent the diam-
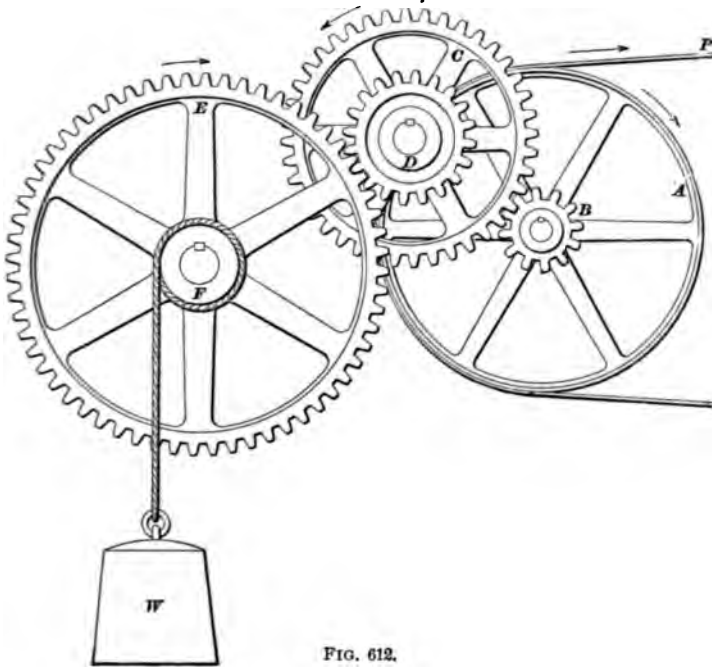
FIG. 612.

eters of the driven wheels and d_1, d_2, d_3, etc., the diameters
of the drivers, we have the following

Rule.—*The continued product of the power and the radii
of the driven wheels equals the continued product of the
weight, the radius of the drum that moves the weight, and
the radii of the drivers.*

This rule gives rise to the following formulas:

$$P = \frac{W \times d_1 \times d_2 \times d_3 \times \ldots}{D_1 \times D_2 \times D_3 \times \ldots}. \qquad \textbf{(100.)}$$

$$W = \frac{P \times D_1 \times D_2 \times D_3 \times \ldots}{d_1 \times d_2 \times d_3 \times \ldots}. \qquad \textbf{(101.)}$$

Example.—The radius of the pulley A is 20 inches, of C, 15 inches,
and of E, 24 inches; and the radius of the drum F is 4 inches, of the
pinion D, 5 inches, and of the pinion B, 4 inches. How great a weight
will a force of 1 pound at P raise?

SOLUTION.—Using formula **101,** we have

$$W = \frac{1 \times 20 \times 15 \times 24}{4 \times 5 \times 4} = \frac{7,200}{80} = 90 \text{ lb.} \quad \textbf{Ans.}$$

If the weight W were raised 1 inch, P would fall 90 inches, or P would have to move 90 inches to raise W 1 inch. *Whenever there is a gain in power without a corresponding increase in the initial force, there is a loss in speed;* this is true of any machine.

In the last example, if P were to move the entire 90 inches in one second, W would move only 1 inch in one second.

1868. Instead of using the diameter or radius of a gear, the number of teeth may be used when computing the weight which can be raised, or the velocity, as in the last example.

EXAMPLE.—Assume that the radius of the pulley A, Fig. 612, is 40 inches, and that of F is 12 inches. The number of teeth in B is 9; in c, 27; in D, 12, and in E, 36. If the weight to be lifted is 1,800 pounds, how great a force at P is it necessary to apply to the belt?

SOLUTION.—Let P represent the force (power); then, by formula **100,**

$$P = \frac{1,800 \times 12 \times 9 \times 12}{40 \times 27 \times 36} = \frac{2,332,800}{38,880} = 60 \text{ lb.} \quad \textbf{Ans.}$$

GEAR-WHEELS.

1869. A wheel that is provided with teeth to mesh with similar teeth upon another wheel is called a **gear-wheel, or gear.** In Fig. 613 is shown a **spur-gear.** On spur-gears, the teeth are always parallel to the axis of the wheel, or to its shaft.

1870. In Fig. 614 is shown a pair of **bevel-gears** in mesh, of which one is smaller than the other. When both are of the same diameter, they are called **miter-gears.**

FIG. 613.

In Fig. 615 is shown a pair

of miter-gears in mesh.   It is obvious that the angle which
the teeth of these gears
make with the axis of
the shaft must be 45°.

1871. In Fig. 616
is shown a revolving
screw, or worm, as it
is called, in gear; it is
used to transmit mo-
tion from one shaft to
another at right angles
to it.

FIG. 614.

As the worm is nothing else than a screw, each revolution
given to the worm will rotate the worm-wheel a distance
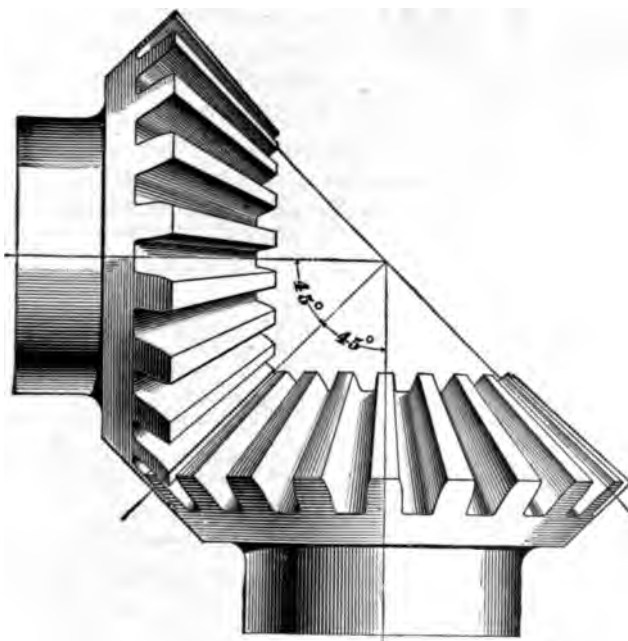
FIG. 615.

equal to its pitch; consequently, if there are 40 teeth in the
worm-wheel, a single-threaded worm will have to make 40
revolutions in order to turn the wheel once.

1872. In Fig. 617 is shown a section of a rack and pinion, both having epicycloidal teeth. The arc *C C* repre-sents part of the **pitch circle;** it is on the pitch circle that all the teeth are laid out. The diameter of a gear or worm-wheel is always taken as the diameter of this circle, unless otherwise specially stated as "diameter over all," or "diam-eter at the root," etc.

The **pitch** of the teeth of the gear-wheel is the distance from the edge of one tooth to the corresponding edge of the fol-lowing tooth measured on the pitch circle; it is marked *pitch* in the figure.

FIG. 616.

The length of the tooth of a gear-wheel is .7 of its pitch, .4 of it, called the **root,** being below or within the pitch circle, and .3 of it, called the **addendum,** being above or
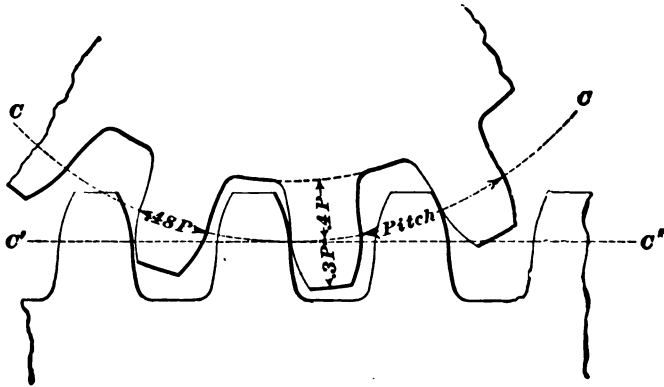
FIG. 617.

without the pitch circle. Thus, if the pitch of the teeth of a gear-wheel is 2 inches, the length of the teeth below the pitch circle is 2 × .4 = .8 of an inch; and the length of

the teeth above the pitch circle is $2 \times .3 = .6$ of an inch. Consequently, we have only to multiply the pitch by .4 to obtain the length of the teeth below the pitch circle, and by .3 to obtain the length of the teeth above the pitch circle. The thickness of the teeth of a cast gear-wheel equals $.48 \times P$, that is, .48 of the pitch; therefore, the thickness of the above teeth is $.48 \times 2$, or .96 of an inch.

A rack may be considered as a gear-wheel rolled out so as to make the pitch circle a straight line, as $C' C''$. The teeth of racks are proportioned by the same rules as those of gear-wheels.

1873. For the purpose of calculating the pitch, diameter, number of teeth, etc., of gear-wheels, the following rules are given:

To find the pitch diameter of a gear-wheel in inches, when the pitch and number of teeth are given:

Rule.—*The pitch diameter is equal to the product of the pitch and number of teeth, divided by 3.1416.*

Let $P =$ pitch;
$T =$ number of teeth;
$D =$ pitch diameter of the wheel.

Then, $$D = \frac{P T}{3.1416}. \qquad \textbf{(102.)}$$

EXAMPLE.—What is the diameter of the pitch circle of a gear-wheel which has 75 teeth, and whose pitch is 1.675 inches?

SOLUTION.—Substituting in formula **102**, we have

$$D = \frac{1.675 \times 75}{3.1416} = 40 \text{ in. Ans.}$$

1874. To find the number of teeth in a gear-wheel, when the diameter and pitch are given:

Rule.—*The number of teeth is equal to the product of 3.1416 and the diameter, divided by the pitch.*

That is, $$T = \frac{3.1416\, D}{P}. \qquad \textbf{(103.)}$$

24—4

EXAMPLE.—The diameter of a gear-wheel is 40 inches, and the pitch of the teeth is 1.675 inches; how many teeth are there in the wheel?

SOLUTION.—Substituting in formula **103**, we have

$$T = \frac{3.1416 \times 40}{1.675} = 75 \text{ teeth. Ans.}$$

1875. To find the pitch of a gear-wheel, when the diameter and the number of teeth are given:

Rule.—*The pitch of the teeth is equal to the product of 3.1416 and the diameter, divided by the number of teeth.*

That is, $P = \dfrac{3.1416\,D}{T}.$ **(104.)**

EXAMPLE.—The diameter of a gear-wheel is 40 inches, and it has 75 teeth; what is the pitch of the teeth?

SOLUTION.—Applying formula **104**, we have

$$P = \frac{3.1416 \times 40}{75} = 1.675 \text{ in. Ans.}$$

1876. The forms of teeth used in ordinary practice are the epicycloidal and involute.

Fig. 617 shows the epicycloidal form, which is composed of two different curves; namely, that curve from the pitch
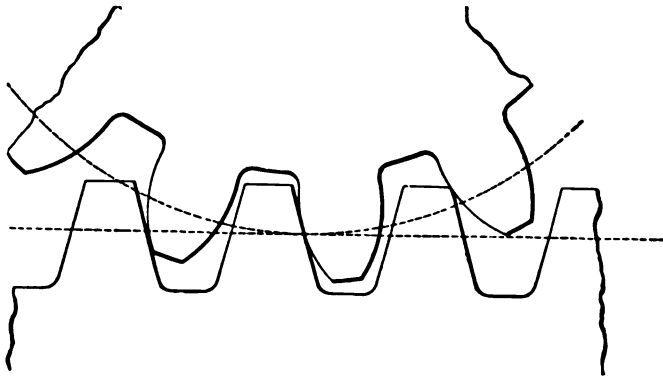
FIG. 618.

circle to the top of the tooth is an epicycloid, and that from the pitch circle to the bottom of the tooth is a hypocycloid.

In gear-wheels where this form of tooth is employed, their pitch circles must run tangent to one another.

1877. In Fig. 618 is shown the involute form of teeth, or teeth having but one curve. The outlines of the teeth shown in the rack are formed of straight lines.

Involute teeth have two great advantages over epicycloidal teeth: (1) They are stronger for the same pitch, as they are thicker at the root. (2) They may be spread apart so that their pitch circles do not run tangent to one another without practically affecting the perfect action of the teeth.

1878. To calculate the number of teeth or speed of one of two gear-wheels which are to gear together:

Let R = number of revolutions per minute of the driver;
   r = number of revolutions per minute of the driven;
   T = number of teeth in the driver;
   t = number of teeth in the driven.

Rule.—*The number of teeth in the driver equals the product of the number of teeth and number of revolutions of the driven, divided by the number of revolutions of the driver.*

That is, $T = \dfrac{t\,r}{R}$.   **(105.)**

EXAMPLE.—The driven has 27 teeth, and will make 66 revolutions per minute; if the driver makes 99 revolutions per minute, how many teeth are there in the driver?

SOLUTION.—Substituting in formula **105**, we have

$$T = \frac{27 \times 66}{99} = 18 \text{ teeth.} \quad \text{Ans.}$$

1879. The number of revolutions per minute of the driver and driven, and the number of teeth in the driver being given, to find the number of teeth in the driven:

Rule.—*The number of teeth in the driven is equal to the product of the number of teeth and revolutions per minute of the driver, divided by the number of revolutions per minute of the driven.*

That is, $t = \dfrac{T\,R}{r}$.   **(106.)**

EXAMPLE.—The driver has 24 teeth, and makes 99 revolutions per minute, and the driven must make 66 revolutions per minute; how many teeth must there be in the driven?

SOLUTION.—Substituting in formula **106**, we have

$$t = \frac{24 \times 99}{66} = 36 \text{ teeth. Ans.}$$

1880. The number of teeth in the driver and driven, and the number of revolutions per minute of the driver being given, to find the number of revolutions per minute of the driven:

Rule.—*The number of revolutions per minute of the driven is equal to the product of the number of teeth and number of revolutions of the driver, divided by the number of teeth of the driven.*

That is, $r = \dfrac{T R}{t}.$ **(107.)**

EXAMPLE.—There are 18 teeth in the driver, and it makes 60 revolutions per minute; how many revolutions per minute will the driven make if it has 30 teeth?

SOLUTION.—Applying formula **107**, we have

$$r = \frac{18 \times 60}{30} = 36 \text{ R. P. M. Ans.}$$

1881. The number of teeth in the driver and driven, and the number of revolutions per minute of the driven being given, to find the number of revolutions per minute of the driver:

Rule.—*The number of revolutions of the driver is equal to the product of the number of teeth and revolutions of the driven, divided by the number of teeth of the driver.*

That is, $R = \dfrac{t r}{T}.$ **(108.)**

EXAMPLE.—If there are 42 teeth in the driven, and if it makes 66 revolutions per minute, how many revolutions per minute will the driver make if it has 18 teeth?

SOLUTION.—Using formula **108**, we have

$$R = \frac{42 \times 66}{18} = 154 \text{ R. P. M. Ans.}$$

EXAMPLE.—In Fig. 619, the crank-shaft makes 60 revolutions per minute; the governor pulley is 4 inches in diameter, and the bevel-

gear on the governor pulley-shaft has 19 teeth; the bevel-gear which meshes with it and drives the governor has 80 teeth. The governor is to make 95 revolutions per minute; what should be the size of the pulley on the crank-shaft?

SOLUTION.—First determine the number of revolutions of the 4-inch pulley in order that the governor shall turn 95 times per minute. Applying formula **108**, we have $R = \dfrac{80 \times 95}{19} = 150$ revolutions of gear on pulley-shaft = revolutions of governor pulley. Now, applying formula **96,** we have $D = \dfrac{4 \times 150}{60} = 10$ in. = diameter of the pulley on the crank-shaft. Ans.

EXAMPLE.—In Fig. 619, the fly-wheel is 8 feet in diameter and drives a 5-foot pulley on the main shaft. A 14-inch pulley on the main shaft drives a 16-inch pulley on the countershaft. A 12-inch pulley on the
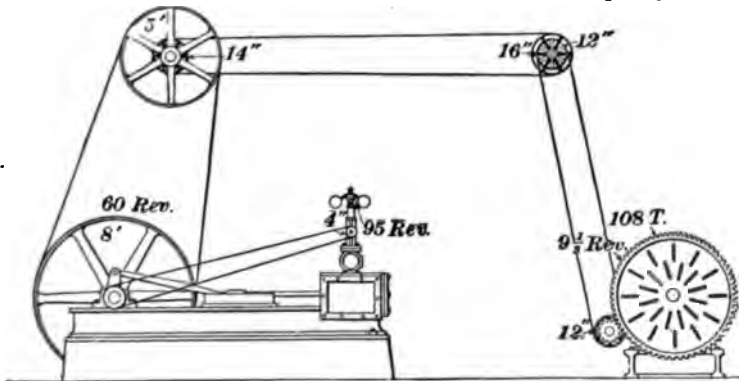
FIG. 619.

countershaft drives a 12-inch pulley on a shaft on which is a pinion that meshes into a large gear, attached to the face-plate of a large lathe, which has 108 teeth. How many teeth must the pinion have in order that the face-plate may make 9¼ revolutions per minute?

SOLUTION.—Applying formula **98,** to find the revolutions per minute of the main shaft, $n = \dfrac{8 \times 60}{5} = 96$ R. P. M. Applying formula **98** again to find the revolutions of the countershaft, $n = \dfrac{14 \times 96}{16} = 84$ R. P. M.; and again to find revolutions of the pulley which turns the small gear, $n = \dfrac{12 \times 84}{12} = 84$ R. P. M. Applying formula **105,** we have $T = \dfrac{108 \times 9\frac{1}{4}}{84} = 12$ teeth in pinion or driver. Ans.

1882. Horsepower of Gears.—To find the horse-power which can be safely transmitted by gears whose face, or breadth of tooth, is from $2\frac{1}{2}$ to 3 times their pitch:

Rule.—*The horsepower which can be safely transmitted equals the continued product of the square of the pitch, the velocity in feet per minute, and .01.*

Let p = the pitch;

   s = circumferential speed of a point on the pitch circle in feet per minute.

Then,   H. P. = $.01 \, s \, p^2$.   **(109.)**

EXAMPLE.—What horsepower can be safely transmitted by a gear whose pitch diameter is 66.84 in., pitch $1\frac{3}{4}$ in., and which makes 60 R. P. M.?

SOLUTION.—The velocity which is to be used when applying formula **108** is the circumferential speed of a point on the pitch circle. Hence, $66.84 \times 3.1416 = 209.98$ in. = circumference of pitch circle = $\frac{209.98}{12}$ ft. $\frac{209.98}{12} \times 60 = 1,049.9$ = velocity in ft. per min.

Now, applying formula **109,** H. P. = $.01 \times 1,049.9 \times 1.75^2 = 32.15$ horsepower. Ans.

1883. When measuring bevel-gears, the diameter of the largest pitch circle should be taken, as D, Fig. 620.
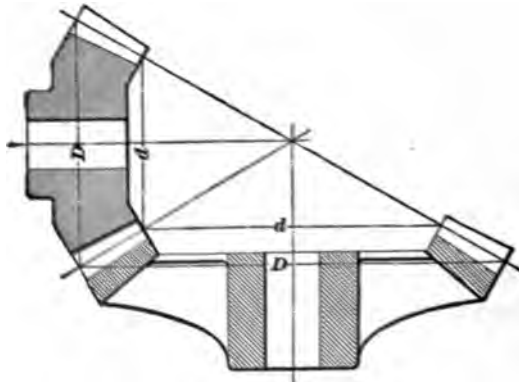
FIG. 620.

When calculating their horsepower, use the small or inner diameter, as d, Fig. 620. Either diameter may be used when calculating the revolutions per minute or number of

teeth by formulas **102** to **108,** but if the inner or outer diameter of one gear be used, the corresponding diameter of the other gear which meshes with it must also be used.

EXAMPLES FOR PRACTICE.

1. The driving pulley makes 110 R. P. M., and is 21 inches in diameter; what should be the size of the driven in order to make 385 R. P. M.? Ans. 6 in.

2. The main shaft of a certain shop makes 120 R. P. M. It is desired to have the countershaft make 150 R. P. M. There are on hand pulleys of 16, 24, 28, 35, and 38 inches in diameter. Can two of these be used, or must a new pulley be ordered? Ans. Use the 28-inch and the 35-inch pulley.

3. The pinion (driver) makes 174 R. P. M. and follower makes 24 R. P. M.; how many teeth must the pinion have if the follower has 87 teeth? Ans. 12 teeth.

4. If an engine fly-wheel is 66 inches in diameter and makes 160 R. P. M., what must be the diameter of the pulley on the main shaft to make 128 R. P. M.? Ans. 82½ in.

5. What is the pitch diameter of a gear whose pitch is 1¼ inches and has 28 teeth? Ans. 11.14 in.

6. How many teeth are there in a gear whose pitch is .7854 inch and which is 23 inches in diameter? Ans. 92 teeth.

7. What is the pitch of a gear whose diameter is 20.372 inches and which has 128 teeth? Ans. ½ in.

8. In a train of gears the drivers have 16, 30, 24, and 18 teeth, respectively; the followers have 12, 24, 36, and 40 teeth, respectively. If the first driver makes 80 R. P. M., how many R. P. M. will the last follower make? Ans. 40 R. P. M.

9. What horsepower can be safely transmitted by a gear whose pitch is 2¼′, pitch diameter 44.66″, and which makes 80 R. P. M.? Ans. 42.24 H. P.

THE INCLINED PLANE AND WEDGE.

1884. An **inclined plane** is a slope, or a flat surface, making an angle with a horizontal line.

Three cases may arise in practice with the inclined plane:

1. When the power acts parallel to the plane, as in Fig. 621.
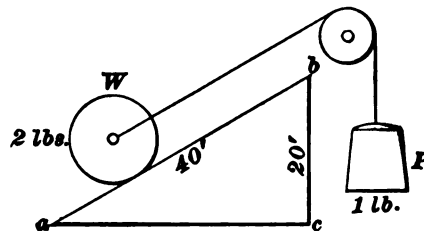
FIG. 621.

2. When the power acts parallel to the base, as in Fig. 622.
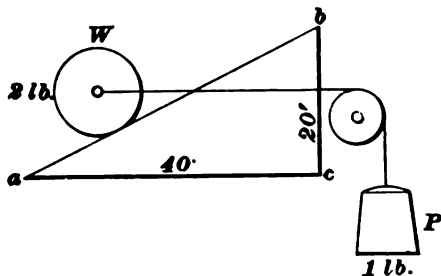
3. When the power acts at an angle to the plane, or to the base, as in Fig. 623.

1885. In Fig. 621, the relation existing between the power and the weight is easily found. The weight ascends a distance equal to *c b*, or the height of the inclined plane, while the

FIG. 622.

power descends through a distance equal to *a b*, or the length of the inclined plane. Therefore, the *power multiplied by the length of the inclined plane equals the weight multiplied by the height of the inclined plane.* Hence, if the length *a b* = 40 feet, and the height *c b* = 20 feet,
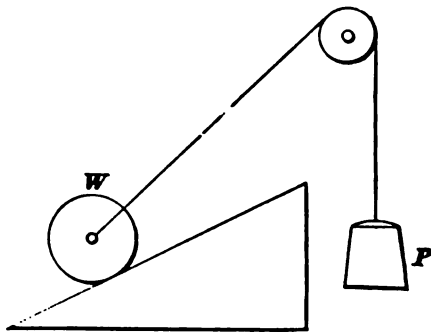
FIG. 621.

$W \times 20 = P \times 40$, or 1 pound at *P* will balance 2 pounds at *W*.

In Fig. 622, the power is supposed to act parallel to the base, for any position of *W*; therefore, while *W* is moving from the level *a c* to *b*, or through the height *c b* of the inclined plane, *P* will move through a distance equal to the length of the base *a c*. Hence, when the power acts parallel to the base, $W \times$ *height of the inclined plane* = $P \times$ *length of base.*

If the length of the base is 40 feet, and the height of the inclined plane is 20 feet, $W \times 20 = P \times 40$, and 1 pound at *P* will balance 2 pounds at *W*.

For Fig. 623 no rule can be given. The ratio of the power to the weight must be determined by trigonometry for every position of *W*.

1886. The **wedge** is a movable inclined plane, and is used for moving a great weight a short distance. A common method of moving a heavy body is shown in Fig. 624.
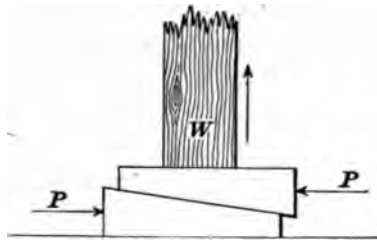
Simultaneous blows of equal force are struck on the heads of the wedges, thus raising the weight W. The laws for wedges are the same as for Case 2 of the inclined plane.

FIG. 624.

THE SCREW.

1887. A **screw** is a cylinder with a helical groove winding around its circumference. This helix is called the *thread* of the screw. The distance that a point on the helix is drawn back or advanced in one turn of the screw is called the *pitch* of the screw.
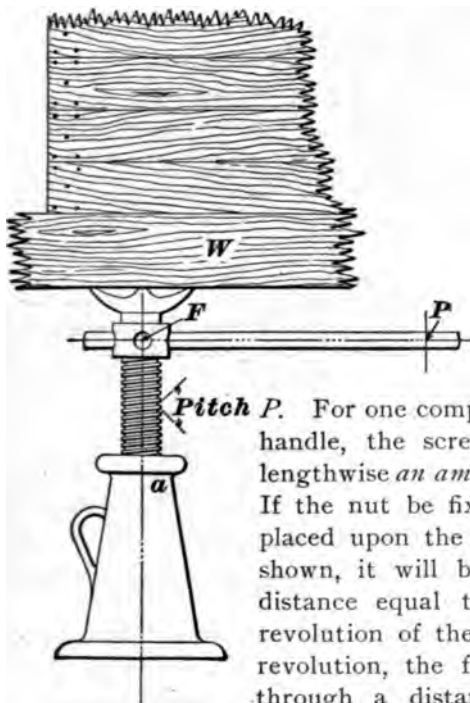
1888. The screw in Fig. 625 is turned in a *nut a*, by means of a force applied at the end of the handle *P*. For one complete revolution of the handle, the screw will be advanced lengthwise *an amount equal to the pitch*. If the nut be fixed, and a weight be placed upon the end of the screw, as shown, it will be raised vertically a distance equal to the pitch, by one revolution of the screw. During this revolution, the force at P will move through a distance equal to the

FIG. 625.

circumference, whose radius is PF.   Hence, $W \times$ pitch of thread $= P \times$ circumference of P.

Let $W =$ weight lifted;
  $P =$ force applied to handle;
  $p =$ pitch of screw;
  $R =$ radius of circle of force P.

Then, $W = \dfrac{6.2832\,PR}{p}.$ **(110.)**

$P = \dfrac{p\,W}{6.2832\,R}.$ **(111.)**

Rule.—*Represent the required force or weight by x; multiply the force by the distance from the center of the screw to the point of the handle where the force is applied; multiply this product by 2 and by 3.1416, and place the result equal to the weight multiplied by the pitch. Divide the product of the known numbers by the number or product of the numbers by which x is multiplied, and the result will be the value of x.*

Single-threaded screws of less than 1-inch pitch are generally classified by the number of threads they have in 1 inch of their length.   In such cases, *one inch divided by the number of threads equals the pitch;* thus, the pitch of a screw that has 8 threads per inch is $\frac{1}{8}$, one of 32 threads per inch is $\frac{1}{32}$, etc.

EXAMPLE.—It is desired to raise a weight by means of a screw having 5 threads per inch.   The force applied is 40 pounds at a distance of 14 inches from the center of the screw; how great a weight can be raised?

SOLUTION.—The pitch is $\frac{1}{5}$ inch.   Using formula **110**,
$$W = \frac{6.2832 \times 40 \times 14}{\frac{1}{5}} = 17,592.96 \text{ lb.   Ans.}$$

1889. Velocity Ratio.—The ratio of the distance that the power moves to the distance which the weight moves on account of the movement of the power is called the **velocity ratio.**

Thus, if the power is moving 12 inches while the weight is moving 1 inch, the velocity ratio is 12 to 1, or 12; that is, P moves 12 times as fast as W.

If the velocity ratio is known, the weight which any machine can raise equals the *power multiplied by the velocity ratio.* If the velocity ratio is 8.7 to 1, or 8.7, $W = 8.7 \times P$, since $W \times 1 = P \times 8.7$.

NOTE.—In all of the preceding cases, including the last, friction has been neglected.

FRICTION.

1890. **Friction** is the resistance that a body meets from the surface on which it moves.

1891. The **ratio** between the *resistance* to the motion of a body due to friction and the *perpendicular* pressure between the surfaces is called the **coefficient of friction.**

If a weight W, as in Fig. 626, rests upon a horizontal plane, and has a cord fastened to it passing over a pulley *a*,
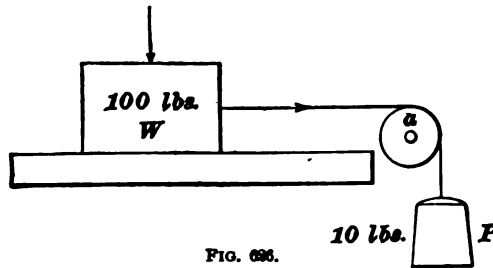
FIG. 626.

from which a weight P is suspended, then, if P is just sufficient to start W, the ratio of P to W, or $\dfrac{P}{W}$, is the *coefficient of friction* between W and the surface it slides upon.

The weight W is the perpendicular pressure, and P is the force necessary to overcome the resistance to the motion of W due to friction.

If $W = 100$ pounds and $P = 10$ pounds, the coefficient of friction for this particular case would be $\dfrac{P}{W} = \dfrac{10}{100} = .1$.

1892. **Laws of Friction :**

I. *Friction is directly proportional to the perpendicular pressure between the two surfaces in contact.*

II. *Friction is independent of the extent of the surfaces in contact when the total perpendicular pressure remains the same.*

III. *Friction increases with the roughness of the surfaces.*

IV. *Friction is greater between surfaces of the same material than between those of different materials.*

V. *Friction is greatest at the beginning of motion.*

VI. *Friction is greater between soft bodies than between hard ones.*

VII. *Rolling friction is less than sliding friction.*

VIII. *Friction is diminished by polishing or lubricating the surfaces.*

1893. Law I shows why the friction is so much greater on journals after they begin to heat than before. The heat causes the journal to expand, thus increasing the pressure between the journal and its bearing, and, consequently, increasing the friction.

Law II states that no matter how small the surface may be which presses against another, if the perpendicular pressure is the same, the friction will be the same. Therefore, large surfaces are used where possible, not to reduce the friction, but to reduce the wear and diminish the liability of heating.

For instance, if the perpendicular pressure between a journal and its bearing is 10,000 pounds, and the coefficient of friction is .2, the amount of friction is $10,000 \times .2 = 2,000$ pounds. Suppose that one-half the area of the surface of the journal is 80 square inches, then the amount of friction for each square inch of bearing is $2,000 \div 80 = 25$ pounds.

If half the area of the surface had been 160 square inches, the friction would have been the same, that is, 2,000 pounds; but the friction per square inch would have been $2,000 \div 160 = 12\frac{1}{2}$ pounds, just one-half as much as before, and the wear and liability to heat would be one-half as great also.

TABLE 31.

COEFFICIENTS OF FRICTION.

Description of Surfaces in Contact.	Disposition of Fibers.	State of the Surfaces.	Coefficient of Friction
Oak on Oak	Parallel	Dry	.48
Oak on Oak	Parallel	Soaped	.16
Wrought Iron on Oak	Parallel	Dry	.62
Wrought Iron on Oak	Parallel	Soaped	.21
Cast Iron on Oak..........	Parallel	Dry	.49
Cast Iron on Oak..........	Parallel	Soaped	.19
Wrought Iron on Cast Iron.	....	Slightly Unctuous	.18
Wrought Iron on Bronze...	....	Slightly Unctuous	.18
Cast Iron on Cast Iron......	....	Slightly Unctuous	.15

1894. The power which is required to raise a weight, or overcome an equal resistance in any machine, is thus always *greater than this weight or resistance divided by the velocity ratio of the machine.*

Thus, if there were no friction, a machine whose velocity ratio were 5 would, by an application of a force of 100 pounds, raise a weight of 500 pounds.

Now, suppose that the friction in the machine is equivalent to the application of a force of 10 pounds; then, it would take a force of 110 pounds to raise the weight of 500 pounds.

If, in the above illustration, friction were neglected, 110 pounds × 5 = 550 pounds, or the weight that 110 pounds would raise; but, owing to the frictional resistance, it only raised 500 pounds. Therefore, we have for the ratio between the two $\frac{500}{550}$ = .91. That is,

$$500 : 550 :: .91 : 1.$$

1895. Efficiency.—This ratio between the weight actually raised and the power multiplied by the velocity ratio is called the **efficiency of the machine.**

For example, if the weight actually raised by a machine, say a screw, is 1,600 pounds, and the power multiplied by the velocity ratio is 2,400 pounds, the efficiency of this machine is $\frac{1,600}{2,400} = .66\frac{2}{3}$, or $66\frac{2}{3}\%$.

EXAMPLE.—In a machine having a combination of pulleys and gears, the velocity ratio of the whole is 9.75. A force of 250 pounds just lifts a weight of 1,626 pounds. What is the efficiency of the machine?

SOLUTION.—Efficiency $= \frac{1,626}{250 \times 9.75} = .6671$, or 66.71%. Ans.

1896. Since the total amount of friction varies with the load, it follows that the efficiency will also vary for different loads.

If the efficiency of a machine is known, the force actually required to raise a given load may be found by dividing the load by the product of the velocity ratio of the machine and the efficiency. Thus, if a certain machine has a velocity ratio of 10.6, and its efficiency is 60%, the force which must actually be applied to raise a load of 840 pounds is $840 \div 10.6 \times .60 = 840 \div 6.36 = 132.1$ pounds, nearly. If there had been no losses through friction, etc., the force required would have been $840 \div 10.6 = 79.25$ pounds, nearly.

If the efficiency is known, the weight which a certain force will raise may be found by multiplying together the force, velocity ratio, and the efficiency. Thus, if a certain machine has a velocity ratio of $6\frac{1}{2}$ and an efficiency of 78%, a force of 140 pounds will raise a weight of $140 \times 6\frac{1}{2} \times .78 = 709.8$ pounds.

When finding the force necessary to overcome the friction, the *perpendicular pressure* on the surface considered must always be taken. Thus, to find the greatest perpendicular pressure on the guides of a steam-engine due to the action of the piston-rod and connecting-rod on the cross-head, multiply the total piston pressure by the length of the crank, and divide by the length of the connecting-rod. This result,

multiplied by the proper coefficient of friction, will give the friction of the cross-head on the guides.

EXAMPLE.—An engine whose piston is 16 inches in diameter carries a steam pressure of 80 pounds per square inch. If the crank is 12 inches long and the connecting-rod is 66 inches long, what is the perpendicular pressure on the guides? The coefficient of friction for this case being 12%, what force will be required to overcome the friction ?

SOLUTION.—Pressure on piston = $16^2 \times .7854 \times 80 = 16,085$ lb. $\dfrac{16,085 \times 12}{66} = 2,924.55$ lb. = perpendicular pressure. Ans. $2,924.55 \times .12 = 350.95$ lb. = force required to overcome the friction. Ans.

EXAMPLES FOR PRACTICE.

1. How great a force must be applied to the free end of the rope of a block and tackle which has four movable pulleys, to raise a weight of 746 pounds ?   Ans. $93\frac{1}{4}$ lb.

2. An inclined plane is 30 feet long and 7 feet high; what force is required to roll a barrel of flour weighing 196 pounds up the plane, friction being neglected ?   Ans. $45.7\frac{1}{4}$ lb.

3. The distance from the axis of a screw to the point on the handle where the force is applied is twelve inches. The screw has 8 threads per inch. What force is necessary to raise a weight of 1,248 pounds ?   Ans. 2.07 lb., nearly.

4. In example 3, what should be the length of the handle to raise a weight of 5,400 pounds by the application of a force of 20 pounds ?   Ans. 5.371 inches, nearly.

5. What is the velocity ratio (a) in example 3 ? (b) in example 4 ?

Ans. $\begin{cases} (a) \text{ 603, nearly.} \\ (b) \text{ 270.} \end{cases}$

6. An engine-piston is 24 inches in diameter. If the steam pressure is 93 pounds per square inch; the length of the connecting-rod, 8 feet 4 inches; the length of crank 20 inches, and coefficient of friction 14%, (a) what is the perpendicular pressure on the guides ? (b) the force required to overcome the friction ?

Ans. $\begin{cases} (a) \text{ 8,414.46 lb.} \\ (b) \text{ 1,178 lb.} \end{cases}$

CENTRIFUGAL FORCE.

1897. If a body be fastened to a string and whirled so as to give it a circular motion, there will be a pull on the string which will be greater or less according as the velocity increases or decreases. The cause of this pull on the string will now be explained.

Suppose that the body is revolved horizontally, so that the action of gravity upon it will always be the same. According to the first law of motion, a body put in motion tends to move in a straight line unless acted upon by some other force, causing a change in the direction. When the body moves in a circle, the force that causes it to move in a circle instead of a straight line is exactly equal to the tension of the string. If the string were cut, the pulling force that draws it away from the straight line would be removed, and the body would then " fly off at a tangent; " that is, it would move in a straight line tangent to the circle, as shown in Fig. 627.
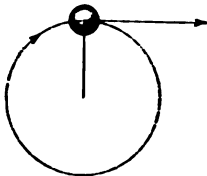
FIG. 627.

1898. Since, according to the third law of motion, every action has an equal and opposite reaction, we call that force which acts as an equal and opposite force to the pull of the string the **centrifugal force,** and it acts *away* from the center of motion.

The other force, or the pull of the string towards the center, is called the **centripetal force,** and it acts *towards* the center of motion. It is evident that these two forces, acting in opposite directions, tend to pull the string apart, and, if the velocity be increased sufficiently, the string will break. It is also evident that no body can revolve without generating centrifugal force.

The value of the centrifugal force, expressed in pounds, of any revolving body is calculated by the following rule:

Rule.—*The centrifugal force is equal to the continued product of .00034, the weight of the body in pounds, the radius in feet (taken as the distance between the center of gravity of the body and the center about which it revolves), and the square of the number of revolutions per minute.*

Let F = centrifugal force in pounds;
W = weight of revolving body in pounds;

> $R =$ radius in feet of circle described by center of gravity of revolving body;
>
> $N =$ revolutions per minute of revolving body.

Then, $F = .00034 \, W R N^2.$ **(112.)**

In calculating the centrifugal force of fly-wheels, it is the usual practice to consider the rim of the wheel only, and not take the arms and hub of the wheel into account. In this case, R would be taken as the *distance between the center of the rim and the center of the shaft.*

EXAMPLE.—A crank-pin weighing 65 pounds revolves in a circle whose radius is 21 inches. The number of revolutions is 180. What is the centrifugal force set up by the pin?

SOLUTION.— 21 in. = 1¾ ft. Using formula **112,**

$$F = .00034 \times 65 \times 1\tfrac{3}{4} \times 180^2 = 1,253.07 \text{ lb. } \textbf{Ans.}$$

SPECIFIC GRAVITY.

1899. The **specific gravity of a body** is the ratio between its weight and the weight of a like volume of water.

Since gases are so much lighter than water, it is usual to take the specific gravity of a gas as the ratio between the weight of a certain volume of the gas and the weight of the same volume of air.

EXAMPLE.—A cubic foot of cast iron weighs 450 pounds; what is its specific gravity, a cubic foot of water weighing 62.5 pounds?

SOLUTION.— $\dfrac{450}{62.5} = 7.2.$ Ans.

1900. The specific gravities of different bodies are given in the tables of Specific Gravities; hence, if it is desired to know the weight of a body that can not be conveniently weighed, *calculate its cubical contents, and multiply the specific gravity of the body by the weight of a like volume of water, remembering that a cubic foot of water weighs 62.5 pounds.*

EXAMPLE.—How much will 3,214 cubic inches of cast iron weigh? Take its specific gravity as 7.21.

24—5

SOLUTION.—Since 1 cubic foot of water weighs 62.5 pounds, 3,214 cubic inches weigh $\frac{3,214}{1,728} \times 62.5 = 116.25$ pounds.

$$116.25 \times 7.21 = 838.16 \text{ pounds. Ans.}$$

EXAMPLE.—What is the weight of a cubic inch of cast iron?

SOLUTION.— $\frac{62.5}{1,728} \times 7.21 = .2608$ pound. Ans.

NOTE.—One cubic foot of pure distilled water at a temperature of 39.2° Fahrenheit weighs 62.42 pounds, but the value usually taken in making calculations is 62½ pounds.

EXAMPLE.—What is the weight in pounds of 7 cubic feet of oxygen?

SOLUTION.—One cubic foot of air weighs .08073 lb. (see table of Specific Gravities), and the specific gravity of oxygen is 1.1056 compared with air; hence, $.08073 \times 1.1056 \times 7 = .62479$ pound, nearly. Ans.

EXAMPLES FOR PRACTICE.

1. The balls of a steam-engine governor each weigh 5 pounds. If they revolve in a circle whose diameter is 14 inches at the rate of 80 revolutions per minute, what is the centrifugal force of each ball?
Ans. 6.347 lb., nearly.

2. If a cubic foot of a certain alloy weighs 678 pounds, what is its specific gravity? Ans. 10.848.

3. What is the weight of (a) 12.4 cubic inches of lead? (b) of steel? (c) of aluminum?
Ans. { (a) 5.0964 lb. (b) 3.5216 lb. (c) 1.116 lb.

4. The specific gravity of an alloy of lead and zinc is 8.26; what is the weight of a cubic foot? Ans. 516.25 lb.

WORK AND ENERGY.

1901. **Work** *is the overcoming of resistance continually occurring along the path of motion.* Mere motion is not work, but if a body in motion constantly overcomes a resistance, it does work.

1902. *The* **measure of work** *is one pound raised vertically one foot*, and is called **one foot-pound.** All work is measured by this standard. A horse going up hill does an amount of work equal to his own weight, plus the weight of the wagon and contents, plus the frictional resistances reduced to an equivalent weight, multiplied by the vertical

height of the hill. Thus, if the horse weighs 1,200 pounds, the wagon and contents 1,200 pounds, and the frictional resistances equal 400 pounds, then, if the vertical height of the hill is 100 feet, the work done is equal to $(1,200 + 1,200 + 400) \times 100 = 280,000$ foot-pounds.

Rule.—*To find the work, multiply the force (or resistance) by the distance through which it acts. If the work consists in raising a weight, it is equal to the product of the weight multiplied by the vertical height of the lift.*

The *total* amount of work is independent of time, whether it takes one minute or one year in which to do it, but in order to compare the work done by different machines with a common standard, time must be considered. If one machine does a certain amount of work in 10 minutes, and another machine does exactly the same amount of work in 5 minutes, the second machine can do twice as much work as the first in the same time.

1903. The common standard to which all work is reduced is the *horsepower*, which is abbreviated H. P. *One horsepower is equal to 33,000 foot-pounds per minute; in other words, it is the work done in raising 33,000 pounds vertically one foot in one minute, or in raising 1 pound vertically 33,000 feet in one minute, or any combination that will, when multiplied together, give 33,000 foot-pounds in one minute.*

Thus, the work done in raising 110 pounds vertically 5 feet in one second is a horsepower, for, since in one minute there are 60 seconds, $110 \times 5 \times 60 = 33,000$ foot-pounds in one minute.

EXAMPLE.—If the coefficient of friction is .3, how many horsepower will be required to draw a load of 10,000 pounds on a level surface, a distance of one mile in one hour ?

SOLUTION.— $10,000 \times .3 = 3,000$ pounds = the force necessary to overcome the resistance (resistance of the air is neglected). One mile = 5,280 feet; one hour = 60 minutes. Therefore, $\dfrac{5,280}{60} = 88$ feet per minute.

Work done = force multiplied by the space $= 3,000 \times 88 = 264,000$ foot-pounds per minute.

Horsepower $= \dfrac{264,000}{33,000} = 8.$ Ans.

1904.   Energy is a term used to express *the ability of an agent to do work.*   Work can not be done without motion, and the work that a moving body is capable of doing in being brought to rest is called the **kinetic energy** of the body.

Kinetic energy means the actual visible energy of a body in motion.   The work which a moving body is capable of doing in being brought to rest is exactly the same as the kinetic energy developed by it in falling in a vacuum through a height sufficient to give it the same velocity.

Rule.—*The kinetic energy of a moving body in foot-pounds equals its weight in pounds multiplied by the square of its velocity in feet per second, and divided by 64.32.*

Let W = weight of body in pounds;
   v = velocity in feet per second;
   K = kinetic energy in foot-pounds.

Then, $$K = \frac{W v^2}{64.32}.\qquad \textbf{(113.)}$$

If a weight is raised to a certain height, a certain amount of work is done equal to the product of the weight and the vertical height.   If a weight is suspended at a certain height and allowed to fall, it will do the same amount of work in foot-pounds that was required to raise the weight to the height through which it fell.

EXAMPLE.—If a body weighing 25 pounds falls from a height of 100 feet, how much work can it do?

SOLUTION.—Work $= W h = 25 \times 100 = 2,500$ foot-pounds.   **Ans.**

It requires the same amount of work or energy to stop a body in motion within a certain time as it does to give it that velocity in the same time.

EXAMPLE.—A body weighing 50 pounds has a velocity of 100 feet per second; what is its kinetic energy?

SOLUTION.—Applying formula **113**,

$$K = \frac{W v^2}{64.32} = \frac{50 \times 100^2}{64.32} = 7,773.63 \text{ foot-pounds.}\quad \textbf{Ans.}$$

EXAMPLE.—In the last example, how many horsepower will be required to give the body this amount of kinetic energy in 3 seconds?

SOLUTION.— 1 H. P. = 33,000 pounds raised 1 foot in 1 minute.

If 7,773.63 foot-pounds of work are done in 3 seconds, in 1 second there

would be done $\dfrac{7,773.63}{3}$ = 2,591.21 foot-pounds of work. One horse-

power = 33,000 ft.-lb. per min. = 33,000 ÷ 60 = 550 ft.-lb. per sec.

The number of horsepower developed will be $\dfrac{2,591.21}{550}$ = 4.71 H. P.

Ans.

1905. **Potential energy** *is latent energy; it is the energy which a body at rest is capable of giving out under certain conditions.*

If a stone is suspended by a string from a high tower, it has potential energy. If the string is cut, the stone will fall to the ground, and during its fall its potential energy will change into kinetic energy, so that at the instant it strikes the ground its potential energy is wholly changed into kinetic energy.

At a point equal to one-half the height of the fall, the potential and kinetic energies are equal. At the end of the first quarter, the potential energy was $\frac{3}{4}$, and the kinetic energy $\frac{1}{4}$; at the end of the third quarter, the potential energy was $\frac{1}{4}$, and the kinetic energy $\frac{3}{4}$.

A pound of coal has a certain amount of potential energy. When the coal is burned, the potential energy is liberated and changed into kinetic energy in the form of heat. The kinetic energy of the heat changes water into steam, which thus has a certain amount of potential energy. The steam acting on the piston of an engine causes it to move through a certain space, thus overcoming a resistance, changing the potential energy of the steam into kinetic energy, and thus doing work.

Potential energy, then, is the energy stored within a body, which may be liberated and produce motion, thus generating kinetic energy, and enabling work to be done.

1906. The principle of **conservation of energy** teaches that energy, like matter, can never be destroyed. If a clock is put in motion, the potential energy of the spring is changed into kinetic energy of motion, which turns the wheels, thus producing friction.

The friction produces heat, which dissipates into the surrounding air, but still the energy is not destroyed—it merely exists in another form. The potential energy in coal was received from the sun in the form of heat ages ago, and has lain dormant for millions of years.

BELTS.

1907. A **belt** is a flexible connecting-band which drives a pulley by its frictional resistance to slipping at the surface of the pulley. Belts are most commonly made of leather or rubber, and united in long lengths by *cementing*, *riveting*, or *lacing*.

Belts are made *single* and *double*. A **single belt** is one composed of a single thickness of leather; a **double belt** is one composed of two thicknesses of leather cemented and riveted together the whole length of the belt.

1908. To Find the Length of a Belt.—In practice, the necessary length for a belt to pass around pulleys that are already in their position on a shaft is usually obtained by passing a tape-line around the pulleys; the stretch of the tape-line is allowed as that necessary for the belt. The lengths of open-running belts for pulleys not in position can be obtained as follows:

Rule.—*The length of a belt for open-running pulleys equals 3¼ times one-half the sum of the diameters of the pulleys plus 2 times the distance between the centers of the shafts.*

Let D = diameter of one pulley;
D_1 = diameter of other pulley;
L = distance between the centers of the shafts;
B = length of the belt.

Then, $B = 3\frac{1}{4}\left(\dfrac{D + D_1}{2}\right) + 2L.$ **(114.)**

EXAMPLE.—The distance between the centers of two shafts is 9 feet 7 inches; the diameter of the large pulley is 36 inches, and the diameter of the small one is 14 inches; what is the necessary length of the belt?

SOLUTION.—Substituting in formula **114,** we have, since **9 feet 7 inches = 115 inches,**

$$B = 3\tfrac{1}{4}\left(\frac{36 + 14}{2}\right) + 2 \times 115 = 311\tfrac{1}{4} \text{ in., or 25 ft. } 11\tfrac{1}{4} \text{ in.   Ans.}$$

1909. To find the width of a single leather belt that will transmit any given horsepower when equal pulleys are used:

Rule.—*The width of the belt in inches equals 800 times the horsepower to be transmitted divided by the speed of the belt in feet per minute.*

Let W = width of single belt in inches;

H = horsepower to be transmitted;

S = speed of belt in feet per minute.

Then,
$$W = \frac{800\,H}{S}. \qquad \textbf{(115.)}$$

EXAMPLE.—What width of single leather belt is required to transmit 20 horsepower when equal pulleys are used and the speed is 1,600 feet per minute?

SOLUTION.—Substituting in formula **115,**

$$W = \frac{800 \times 20}{1,600} = 10 \text{ inches.   Ans.}$$

1910. To find the number of horsepower that a single leather belt will transmit, its width and speed being given:

Rule.—*The number of horsepower equals the product of the width in inches and the speed in feet per minute divided by 800.*

Or,
$$H = \frac{WS}{800}. \qquad \textbf{(116.)}$$

EXAMPLE.—If a 16-inch single leather belt is to be run at a speed of 700 feet per minute, what horsepower will it transmit?

SOLUTION.—Substituting in formula **116,** we have

$$H = \frac{16 \times 700}{800} = 14 \text{ horsepower.   Ans.}$$

When the pulleys are of different diameter, the arc of contact must be considered. To find the number of degrees in the arc of contact, *multiply the length of belt in contact on the smaller pulley by 360, and divide the product by the*

circumference of the pulley, calculating the result to the nearest whole number. The quotient is the arc of contact.

Having found the arc of contact, *subtract it from 180° and multiply the result by 3. Add this last result to 800; the number thus obtained should be used instead of 800 in formulas* **115** *and* **116.**

EXAMPLE.—What should be the width of a single leather belt to transmit 25.24 horsepower at a speed of 1,500 feet per minute, the diameter of the smaller pulley being 24", and the belt having 80° of its length in contact with it?

SOLUTION.—Arc of contact $= \dfrac{30 \times 360}{24 \times 3.1416} = 143°$. $(180 - 143) \times 3 =$ 111. $800 - 111 = 911$. Using formula **115,** and 911 instead of 800,

$$W = \frac{911 \times 25.24}{1.500} = 15.33", \text{ say } 15\tfrac{1}{4}'. \quad \text{Ans.}$$

1911. To find the width of a double belt that will transmit the same horsepower as a given single belt, let W_1 represent the width of the double belt; then,

Rule.—*Multiply the width of a single belt that will transmit the same horsepower by ⅔.*

Or, $W_1 = \tfrac{2}{3} W.$ **(117.)**

EXAMPLE.—If a single leather belt is 15" in width and transmits 21.818 horsepower, what must be the width of a double belt to transmit the same horsepower?

SOLUTION.—Applying formula **117,**

     $W_1 = 15 \times \tfrac{2}{3} = 10$ in. $=$ width of double belt.   Ans.

1912. Lacing Belts. —Many good methods of fastening the ends of belts are employed, but lacing is generally used, as it is flexible like the belt, and runs noiselessly over the pulleys.

When punching a belt for lacing, use an oval punch, the long diameter of the hole to be parallel with the side of the belt.

In a 3-inch belt, there should be four holes in each end, two in each row. In a 6-inch belt, seven holes, four in the row nearest the end. A 10-inch belt should have nine holes, five in the row nearest the end. The edges of the holes

should not be nearer than ¼ of an inch from the sides, and ⅛ of an inch from the ends of the belt. The second row should be at least 1¾ inches from the end.

Always begin to lace from the center of the belt, and take care to get the ends exactly in line. The lacing should not be crossed on the side of the belt that runs next to the pulley. Always run the hair side of the belt next to the pulley.

EXAMPLES FOR PRACTICE.

1. How many foot-pounds of work are required to overcome for 7 minutes the friction of the cross-head of an engine which has a stroke of 4 feet and makes 160 strokes per minute, if the coefficient of friction is 8% and the average perpendicular pressure is 12,460 pounds?

Ans. 4,465,664 ft.-lb.

2. In the above example, what horsepower is required?

Ans. 19.332 H. P.

3. A cannon-ball weighing 500 pounds is fired with a velocity of 1,600 feet per second; what is its kinetic energy?

Ans. 19,900,497.5 ft.-lb.

4. An open belt drives two pulleys which are respectively 42 inches and 20 inches in diameter and 23 feet apart between their centers; what should be the length of the belt? Ans. 652¼ in., or 54 ft. 4¼ in.

5. What width of single leather belting, which has 2 feet 9 inches contact on the small pulley, is required to transmit 10 horsepower at a speed of 1,500 feet per min.? Give width to nearest half inch. Diameter of small pulley, 26 inches. Ans. 6 in.

6. What should be the width of the main belt of a steam-engine to transmit 120 horsepower? The engine runs at 80 R. P. M., the band wheel is 8 feet in diameter, the belt is double and has a contact of 6 feet on the smaller pulley, which is 5 feet in diameter. Take the speed of the belt the same as that of a point on the circumference of the band-wheel. Ans. 36¼ in.

7. A 26-inch double belt runs at a speed of 2,830 feet per min. and has a contact of 5 feet on the smaller pulley; what horsepower is it transmitting? Diameter of small pulley is 48 inches.

Ans. 121.15 H. P.

MECHANICS.

(PART 2.)

THE COMPOSITION OF FORCES.

1913. When two forces act upon a body at the same time but at different angles, their final result may be obtained as follows:

In Fig. 628, let *A* be the common *point of application* of the two forces, and let *A B* and *A C* represent the *magnitude* and *direction* of the forces. According to the second law of motion, the final effect of the movement due to these two forces would be the same, whether they acted singly or together. Suppose that the line *A B* represents the distance that the force *A B* would cause the body to move; similarly, that *A C* represents the distance which the force *A C* would cause the body to move when both forces were acting separately. The force *A B*, acting alone, would carry the body to *B*; if the force *A C* were now to act upon the body, it would carry it along the line *B D*, parallel to *A C*, to a point *D*, at a distance from *B* equal to *A C*. Join *C* and *D*; then, *C D* is parallel to *A B*, and *A B D C* is a parallelogram. Draw the diagonal *A D*. According to the second law of motion, the body will stop at *D*, whether the forces act separately or together, but if they act together, the path of the body will be along *A D*, the diagonal of the parallelogram. Moreover, the length of the line *A D* represents the *magnitude* of a force, which, acting at *A* in the *direction A D*, would
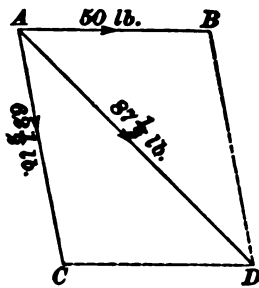
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cause the body to move from A to D; in other words, A D, measured to the same scale as A B and A C, represents in *magnitude* and *direction* the combined effect of the two forces A B and A C.

This line A D is called the **resultant.** Suppose that the scale used was 50 pounds to the inch; then, if A $B=50$ pounds, and A $C = 62\frac{1}{2}$ pounds, the length of A B would be $\frac{50}{50} = 1$ inch, and the length of A C would be $\frac{62.5}{50} = 1\frac{1}{4}$ inches. If A D, or the *resultant*, measures $1\frac{3}{4}$ inches, its *magnitude* would be $1\frac{3}{4} \times 50 = 87\frac{1}{2}$ pounds.

Therefore, a force of $87\frac{1}{2}$ pounds acting upon a body at A in the direction $A \cdot D$ will produce the same result as the combined effects of a force of 50 pounds acting in the direction A B, and a force of $62\frac{1}{2}$ pounds acting in the direction A C.

1914. The above method of finding the resulting action of two forces acting upon a body at a common point is correct, whatever may be their direction and magnitudes. Hence, to find the **resultant** of two forces when their common point of application, their direction, and magnitudes are known:

Rule.—*Assume a point, and draw two lines parallel to the directions of the lines of action of the two forces. With any convenient scale, measure off from the point of intersection (common point of application) distances corresponding to the magnitudes of the respective forces, and complete the parallelogram. From the common point of application, draw the diagonal of the parallelogram; this diagonal will be the resultant, and its direction will be away from the point of application. Its magnitude should be measured with the same scale that was used to measure the two forces.*

This method is called the **graphical method of the parallelogram of forces.**

1915. The principle of the parallelogram of forces is clearly shown in Fig. 629. A B D C is a wooden frame, jointed to allow motion at its four corners. The length

$A\,B$ equals $C\,D$; that of $A\,C$ equals $B\,D$, and the corresponding adjacent sides are in the ratio of 2 to 3.   Cords pass over the pulleys M and N, carrying weights W and w, of 90 and 60 pounds.   The ratio between the weights equals the ratio of the corresponding adjacent sides.   A weight V of 120 pounds is hung from the corner A.

When the frame comes to rest, the sides $A\,B$ and $A\,C$ lie
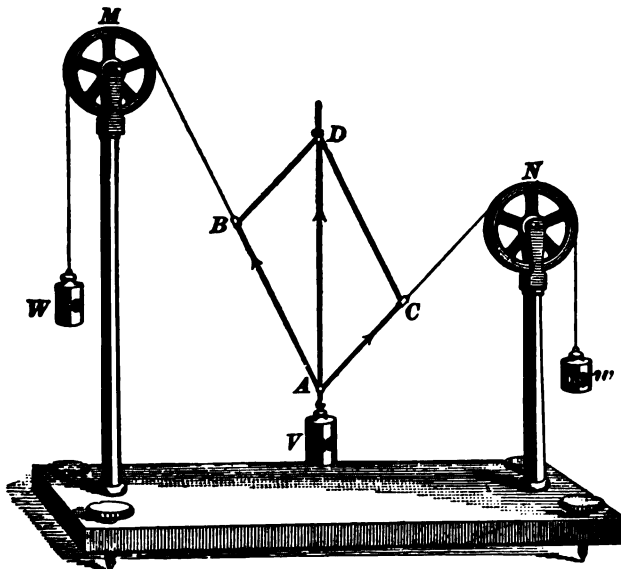
FIG. 629.

in the direction of the cords.   These sides $A\,B$ and $A\,C$ are accurate graphic representations of the two forces acting upon the point A.   It will be found that the diagonal $A\,D$ is vertical, and twice as long as $A\,C$; hence, since $A\,C$ represents a force of 60 pounds, $A\,D$ will represent a force of 2×60, or 120 pounds.

Thus, we see that the line $A\,D$ represents the *resultant* of the two forces $A\,B$ and $A\,C$; in other words, it represents the resultant of the two weights W and w.   This resultant is equal and opposite to the vertical force, which is due to the weight of V.

Satisfactory results of this kind will be secured when we have the proportion

$$A B : A C = W : w.$$

EXAMPLE.—If two forces act upon a body at a common point, both acting away from the body, and the angle between them is 80°, what is the value of the resultant, the magnitude of the two forces being 60 pounds and 90 pounds, respectively ?

SOLUTION.—Draw two indefinite lines having an angle of 80° between them. With any convenient scale, say 10 pounds to the inch, measure off $A B = 60 \div 10 = 6$ inches, and $A C = 90 \div 10 = 9$ inches.
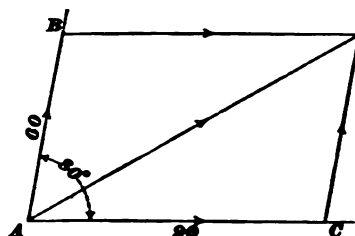
Through B, draw $B D$ parallel to $A C$, and through C, draw $C D$ parallel to $A B$, intersecting at D. Then, draw $A D$, and $A D$ will be the *resultant;* its *direction* is towards the point D, as shown by the arrow.

FIG. 630.

Measuring $A D$ we find that its length = 11.7 inches. Hence, $11.7 \times 10 = 117$ pounds. Ans.

CAUTION.—In solving problems by the graphical method, *use as large a scale as possible.* More accurate results are then obtained.

1916. The above example might also have been solved by the method called the **triangle of forces,** which is as follows:

In Fig. 630, suppose that the two forces acted separately, first from A to B, and then from B to D, in the direction of the arrows.

Draw $A D$; then $A D$ is the *resultant* of the forces $A B$ and $A C$, since $B D = A C$; but $A D$ is a side of the triangle $A B D$. It will also be noticed that the direction of $A D$ is *opposed* to that of $A B$ and $B D$; hence, to find the **resultant** of two forces acting upon a body at a common point, by the method of triangle of forces:

Rule.—*Draw the lines of action of the two forces as if each force acted separately, the lengths of the lines being proportional to the magnitude of the forces. Join the extremities of the two lines by a straight line,* **and it will**

be the resultant; its direction will be opposite to that of the two forces.

NOTE.—When we speak of the resultant being opposed in direction to the other forces around the polygon, we mean that, starting from the point where we began to draw the polygon, and tracing each line in succession, the pencil will have the same general direction around the polygon as if passing around a circle, from left to right, or from right to left, but that the closing line or resultant must have an *opposite direction*; that is, *the two arrow-heads must point towards the point of intersection of the resultant and the last side.*

1917. EXAMPLE.—Suppose the center of a headwheel is elevated 100 feet above the center of a hoisting-drum, as shown in Fig. 631. The rope from the headwheel to the hoisting-drum makes an angle of 80° with a vertical line, and the weight of the carriage and the load to be hoisted is 5 tons. (1) What force will there be on the shaft of the headwheel ? (2) In what direction will the resultant force act, or what would be the direction in which the headwheel would be thrown if its shaft should break ?

SOLUTION.—In Fig. 631, *A B C* represents the rope and its direction, with one end fastened to load *C.* The other end is passed over head-wheel *B*, and wound around drum *A*. Now, as the rope is held in position by drum *A*, the tension at any point is equal to load *C.* Consequently, there is a force of 5 tons acting in the direction from *B* to *A*, as indicated by the arrow, and a like force acting in the direction from *B* to *C*, as indicated by the arrow. *B C* is assumed to be vertical. If we produce the lines *A B* and *C B* to *d*, *d* is the point of application; thus, we have the point of application, magnitude, and direction of the acting forces. Now, if we use a scale 1 inch = 1 ton, and lay off from *d*, the point of applica-
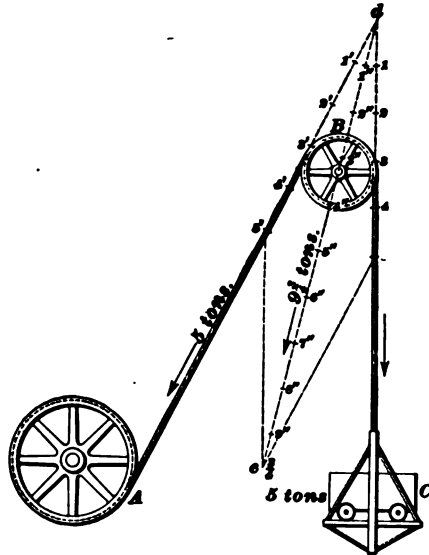
FIG. 631.

tion, five inches or divisions on each component, as *d* to *1'*, *1'* to *2'*, *2'* to *3'*, *3'* to *4'*, *4'* to *5'*, and *d* to *1*, *1* to *2*, *2* to *3*, *3* to *4*, *4* to *5*, each inch or division represents one ton, and, consequently, the five inches

or divisions represent five tons, or the total force of each component. Then, by completing the parallelogram *d 5' e 5*, by drawing line *5' e* parallel to *C B d*, and line *5 e* parallel to *A B d*, we have only to find how many times the resultant *d e* contains the distance *d 1*. If the resultant contains *d 1* seven times, then there is a force of 7 tons on the shaft *B*, acting in the direction *d e*, or if it contains *d 1* ten times, then there is a force of 10 tons on the shaft *B*, and so on. Conscquently, there is one ton for each division we get on the line *d e*. Fig. 631 shows 9¼ such divisions; consequently, there are 9¼ tons on the shaft *B*, acting in the direction *d e*. The above discussion supposes the parts to be at rest.

1918. When three or more forces act upon a body at a given point, their *resultant* may be found by the following rule:

Rule.—*Find the resultant of any two forces; treat this resultant as a single force, and combine it with a third force to find a second resultant. Combine this second resultant with a fourth force, to find a third resultant, etc. After all the forces have been thus combined, the last resultant will be the resultant of all the forces, both in magnitude and direction.*

EXAMPLE.—Find the resultant of all the forces acting on the point *O* in Fig. 632, the length of the lines being proportional to the magnitude of the forces.

SOLUTION.—Draw *O E* parallel and equal to *A O*, and *E F* parallel and equal to *B O*; then, *O F* is the resultant of these two forces, and its
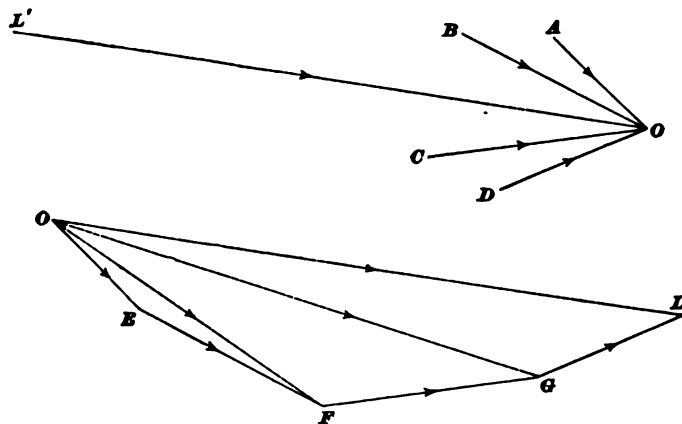
FIG. 632.

direction is from O to F, opposed to OE and EF. Treat OF as if OE and EF did not exist, and draw FG parallel and equal to CO; OG will be the resultant of OF and FG; but OF is the resultant of OE and EF; hence, OG is the resultant of OE, EF, and FG and likewise of AO, BO, and CO. The line FG parallel to CO could not be drawn from the point O to the right of OE, for in that case it would be opposed in direction to OF; but FG must have the same direction as OF, in order that the resultant may be opposed to both OF and FG.

For the same reason, draw GL parallel and equal to DO. Join O and L, and OL will be the *resultant* of all the forces AO, BO, CO, and DO (both in magnitude and direction), acting at the point O. If $L'O$ were drawn parallel and equal to OL, and having the same direction, it would represent the effect produced on the body by the combined action of the forces AO, BO, CO, and DO.

In Fig. 632, it will be noticed that OE, EF, FG, GL, and LO are sides of a polygon $OEFGL$, in which OL, the resultant, is the closing side, and that its direction is opposed to that of all the other sides. This fact is made use of in what is called the **method of the polygon of forces.**

1919. To find the resultant of several forces acting upon a body at the same point:

Rule.—*Through a convenient point on the drawing, draw a line parallel to one of the forces, and having the same direction and magnitude. At the end of this line, draw another line parallel to a second force, and having the same direction and magnitude as this second force; at the end of the second line, draw a line parallel and equal in magnitude and direction to a third force. Thus continue until lines have been drawn parallel and equal in magnitude and direction to all of the forces.*

The straight line joining the free ends of the first and last lines will be the closing sides of the polygon ; mark it opposite in direction to that of the other forces around the polygon, and it will be the resultant of all the forces.

EXAMPLE.—If five forces act upon a body at angles of 60°, 120°, 180°, 240°, and 270°, towards the same point, and their respective magnitudes

are 60, 40, 30, 25, and 20 pounds, find the magnitude and direction of their resultant by the method of polygon of forces.*

SOLUTION.—From a common point O, Fig. 633, draw the lines of action of the forces, making the given angles with a horizontal line through O, and mark them as acting towards O, by means of arrow-heads, as shown. Now, choose some convenient scale, such that the
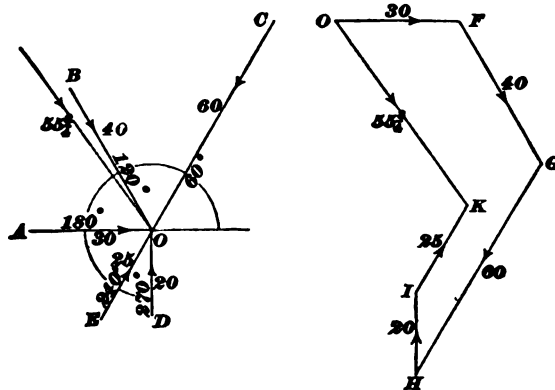
FIG. 633.

whole figure may be drawn in a space of the required size on the draw-ing. Choose any one of the forces, as $A\,O$, and draw $O\,F$ parallel to it, and equal in length to 30 pounds on the scale. It must also act in the same direction as $A\,O$. At F, draw $F\,G$ parallel to $B\,O$, and equal to 40 pounds. In a similar manner, draw $G\,H$, $H\,I$, and $I\,K$ parallel to $C\,O$, $D\,O$, and $E\,O$, and equal to 60, 20, and 25 pounds, respectively. Join O and K by $O\,K$, and $O\,K$ will be the resultant of the combined action of the five forces; its direction is opposite to that of the other forces around the polygon $O\,F\,G\,H\,I\,K$, and its magnitude = 55¾ pounds. Ans.

1920. If the resultant $O\,K$, in Fig. 633, were to act alone upon the body in the direction shown by the arrow-head with a force of 55¾ pounds, it would produce exactly the same effect upon a body as the combined action of the five forces.

If $O\,F$, $F\,G$, $G\,H$, $H\,I$, and $I\,K$ represent the distances and directions that the forces would move the body, if acting

* NOTE.—As stated in Trigonometry, all angles are measured from a horizontal line in a direction opposite to the movement of the hands of a watch (from around the circle to the left), from 1° or less, up to 360°.

separately, $O\ K$ is the direction and distance of movement of the body when all the forces act together.

From what has been said before, it is seen that any number of forces acting on the body at the same point, or having their lines of action pass through the same point, can be replaced by a *single force* (resultant) whose line of action shall pass through that point.

1921. Heretofore it has been assumed that the forces acted upon a single point on the *surface* of the body, but it
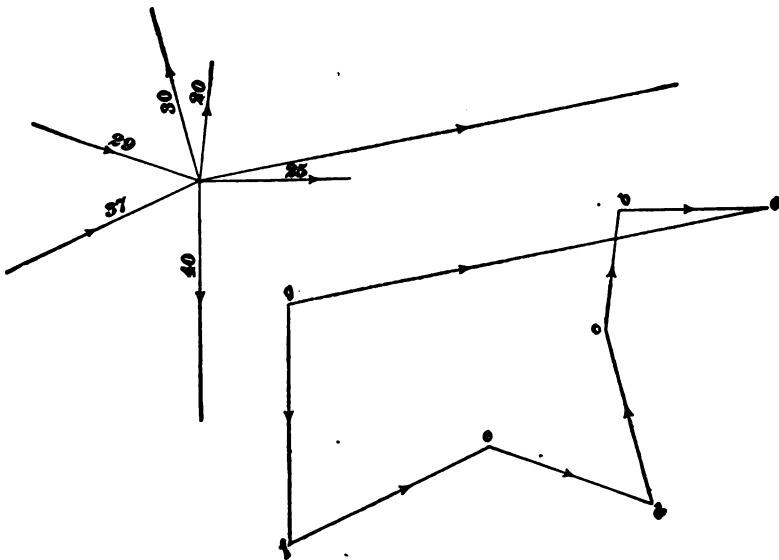
FIG. 634.

will make no difference where they act, so long as the lines of action of all the forces intersect at a *single point* either within or without the body, only so that the resultant can be drawn through the *point of intersection*. If two forces act upon a body in the same straight line and in the same direction, their *resultant* is the *sum of the two forces;* but if they act in opposite directions, their *resultant* is the *difference of the two forces,* and its direction is the same as that of the

greater force.   If they are equal and opposite, the *resultant* is *zero*, or one force just balances the other.

EXAMPLE.—Find the resultant of the forces whose lines of action pass through a single point, as shown in Fig. 684.

SOLUTION.—Take any convenient point g, and draw a line gf, parallel to one of the forces, say the one marked 40, making it equal in length to 40 pounds on the scale, and indicate its direction by the arrow-head.   Take some other force—the one marked 87 will be convenient; the line fe represents this force.   From the point e, draw a line parallel to some other force, say the one marked 29, and make it equal in magnitude and direction to it.   So continue with the other forces, taking care that the general direction around the polygon is not changed.   The last force drawn in the figure is ab, representing the force marked 25.   Join the points a and g; then, ag is the result-ant of all the forces shown in the figure.   Its direction is from g to a, opposed to the general direction of the others around the polygon.   It does not matter in what order the different forces are taken, the result-ant will be the same in magnitude and direction, if the work is done correctly.

THE RESOLUTION OF FORCES.

1922.   Since two forces can be combined to form a single resultant force, we may also treat a single force as if it was the resultant of two forces, whose action upon a body
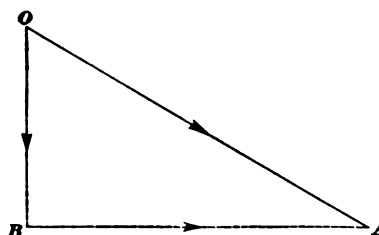
FIG. 635.

will be the same as that of a single force.   Thus, in Fig. 635, the force OA may be resolved into two forces, OB and BA, whose direc-tions are opposed to OA.

If the force OA acts upon a body, moving or at rest upon a horizontal plane, and the resolved force OB is ver-tical, and BA horizontal, OB, measured to the same scale as OA, is the magnitude of that part of OA which pushes the body *downwards*, while BA is the magnitude of that part of the force OA which is exerted in pushing the body in a *horizontal* direction. OB and BA are called the **components** of the force OA, and when these components

are vertical and horizontal, as in the present case, they are called the *vertical component* and the *horizontal component* of the force $O\,A$.

1923. It frequently happens that the position, magnitude, and direction of a certain force is known, and that it is desired to know the effect of the force in some direction other than that in which it acts. Thus, in Fig. 635, suppose that $O\,A$ represents, to some scale, the magnitude, direction, and line of action of a force acting upon a body at A, and that it is desired to know what effect $O\,A$ produces in the direction $B\,A$. Now, $B\,A$, instead of being horizontal, as in the cut, may have any direction. To find the value of the component of $O\,A$ which acts in the direction $B\,A$, we employ the following rule:

Rule.—*From one extremity of the line representing the given force, draw a line parallel to the direction in which it is desired that the component shall act; from the other extremity of the given force, draw a line perpendicular to the component first drawn, and intersecting it. The length of the component, measured from the point of intersection to the intersection of the component with the given force, will be the magnitude of the effect produced by the given force in the required direction.*

Thus, suppose $O\,A$, Fig. 635, represents a force acting upon a body resting upon a horizontal plane, and it is desired to know what *vertical pressure* $O\,A$ produces on the body. Here the desired direction is vertical; hence, from one extremity, as O, draw $O\,B$ parallel to the desired direction (vertical in this case), and, from the other extremity, draw $A\,B$ perpendicular to $O\,B$, and intersecting $O\,B$ at B. Then $O\,B$, when measured to the same scale as $O\,A$, will be the value of the vertical pressure produced by $O\,A$.

EXAMPLE.—If a body weighing 200 pounds rests upon an inclined plane whose angle of inclination to the horizontal is 18°, what force does it exert perpendicular to the plane, and what force does it exert parallel to the plane, tending to slide downwards?

SOLUTION.—Let $A\,B\,C$, Fig. 636, be the plane, the angle A being

equal to 18°, and let W be the weight. Draw a vertical line $FD =$ 200 pounds, to represent the magnitude of the weight. Through F, draw FE parallel to AB, and through D draw DE perpendicular to EF, the two lines intersecting at E. FD is now resolved into two components, one, FE, tending to pull the weight downwards, and the other, ED, acting as a perpendicular pressure on the plane.
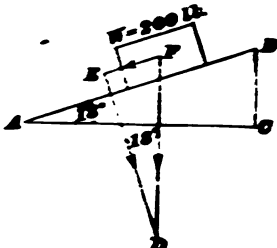
Since FD is perpendicular to AC and ED is perpendicular to AB, the angle $D =$ angle $A = 18°$.

Hence, $FE = 200 \times \sin 18° = 200 \times .30902 = 61.804$ pounds, and $ED = 200 \times \cos 18° = 200 \times .95106 = 190.212$ pounds.

Fig. 636.

$$200 \times \cos 18° = 200 \times .95106 = 190.212 \text{ pounds.}$$

Force parallel to the plane $= 61.804$ pounds.   } Ans.
Force perpendicular to the plane $= 190.212$ pounds. }

1924. EXAMPLE.—In Fig. 637, a body W is shown resting on an inclined plane AB, whose dimensions are marked on the cut; the weight P acts to pull the body up the plane by means of the rope r and pulley p. It is required to find what the weight of P must be in order to start W up the plane. Suppose W weighs 120 pounds, and that friction is neglected. It is also required to find the perpendicular pressure which W exerts against the plane.
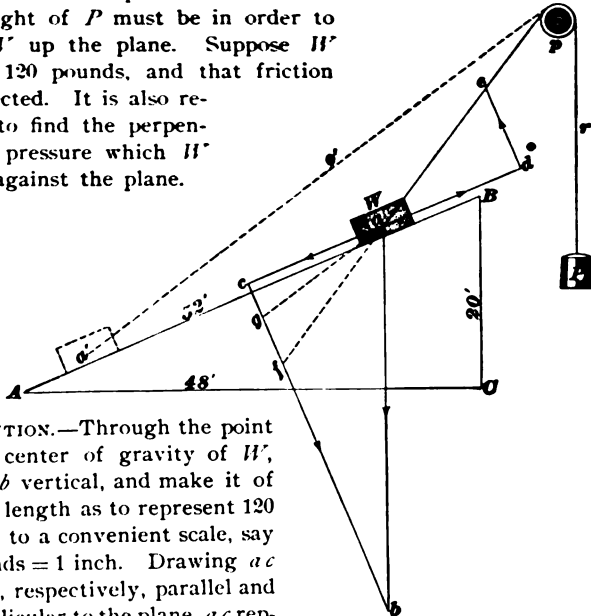
SOLUTION.—Through the point a, the center of gravity of W, draw $a\,b$ vertical, and make it of such a length as to represent 120 pounds to a convenient scale, say 60 pounds $= 1$ inch. Drawing $a\,c$ and $c\,b$, respectively, parallel and perpendicular to the plane, $a\,c$ represents the magnitude of the force

Fig. 637.

which must be exerted parallel to *A B* in order to put the body in
equilibrium, i. e., to balance the force which gravity exerts in pulling
the body down the plane. If the rope *r* were parallel to *A B*, *a c* would
represent the weight of *P*, but, since *r* makes an angle with the plane,
P will not be equal to *a c*. To find what the weight of *P* must be,
draw *a d* parallel to *c r*, but indicate it as acting in the opposite direc-
tion, or from *a* to *d* instead of from *d* to *a*. Now, treat *c d* as though
it were a *component* of the force acting in the rope; i. e., draw *d c* per-
pendicular to *a d*, instead of perpendicular to *c r*. The reason for this
is that if *d c* were drawn perpendicular to *c r*, it could be resolved into
components, one of which would be parallel to *c d*, a result which we
wish to avoid; in other words, we want *d c* perpendicular to the plane.
The line *a c*, measured to the same scale as *c d*, will give the value of
P. Measuring it, its length is .89 inch; hence, *P* = .89 × 60 = 53.4
pounds. Ans.

To determine the perpendicular pressure against the plane, it will be
noticed that *a b* equals the pressure due to gravity. Since *c d* and *d c*
are both perpendicular to *A B*, they are parallel; and since *d c* acts in
the opposite direction to *c d*, the actual pressure against the plane is
given by the difference between *c d* and *d c*. Making *c f* equal to *d c*,
f b represents the perpendicular pressure against the plane when the
force *P* (= *a c*) acts as shown. The length of *f b* is 1.39 inches; hence,
the perpendicular pressure is 1.39 × 60 = 83.4 pounds. Ans.

Since *c a* and *a d* are parallel and equal, and *c f* and *d c* are also
parallel and equal, it follows that *a f* and *a c* must also be parallel and
equal. Consequently, the force *P* might have been found by drawing
a f parallel to the direction in which the pull on the rope acts, and *b f*
perpendicular to the plane *A B*. Thus, suppose that the weight
occupies the position shown by the dotted lines. Then, drawing *a g*
parallel to *a c*, *a g* represents the weight of *P*, and *g b* represents
the perpendicular pressure of the body *W* against the plane. Measur-
ing *a g*, its length is .79 inch; hence, *P* = .79 × 60 = 47.4 pounds.
Measuring *g b*, its length is 1.65 inches; hence, the perpendicular pres-
sure = 1.65 × 60 = 99 pounds.

1925. The results obtained by the graphic method can
be obtained by trigonometry when the inclination of the
plane and the angle the rope makes with the plane for any
position of the weight W_1 are given.

Thus, $a c = a b \times \sin a b c = 120 \times \frac{99}{99} = 46.1538$ pounds.

Assuming the weight w to be in such a position that the
rope r makes an angle of 30° 12′ with the inclined plane, and

since in the triangle $a\,d\,e$ the side $a\,d$ equals the side $c\,a$ in the triangle $a\,b\,c$, we have

$$a\,e = \frac{a\,d}{\cos e\,a\,d} = \frac{46.1538}{.86427} = 53.4 \text{ pounds.} \quad \textbf{Ans.}$$

EXAMPLES FOR PRACTICE.

1. The current in a river which is ½ mile wide has a velocity of 3½ miles an hour. (*a*) What will be the actual distance that a boat will pass over in crossing the river, if the boat is rowed at the rate of 5 miles an hcur ? (*b*) How far down the river will the boat have been carried when it reaches the other side? (*c*) What time will the boat require to cross the river ?

Ans. $\begin{cases} (a) \text{ } \frac{5}{8} \text{ mi.} \\ (b) \text{ } \frac{7}{16} \text{ mi.} \\ (c) \text{ 6 min.} \end{cases}$

2. What force acting parallel to a plane whose inclination is 30° will be required to support a trip of cars whose total weight is 25 tons ?

Ans. 12½ tons.

3. If a driver takes a side-hitch on a trip of cars standing on the turnout, with a mule that pulls with a force of say 400 pounds in a direction making an angle of 45° with the track, what force will tend to move the trip along the track ?   ·   Ans. 282.85 lb.

4. Referring to Fig. 637, what would the angle $e\,a\,d$ become, if $P = 65.271$ pounds ?   Ans. 45°.

5. *The two ends of a rope 7 feet long are attached to the under side of a beam at points 5 feet apart; if a weight of one hundred pounds is firmly attached to the rope at a point 4 feet from one end, what will be the tension in each segment of the rope ?

Ans. $\begin{cases} 60 \text{ lb. tension in long segment.} \\ 80 \text{ lb. tension in short segment.} \end{cases}$

6. What weight can be supported on a plane by a horizontal force of 1,521 pounds, if the ratio of the height to the base is ¾ ?

Ans. 2,028 lb.

7. What force is required (neglecting friction) to roll a barrel of oil weighing 300 pounds into a wagon 3 feet high by means of a plank 14 feet long resting against the wagon ?   Ans. 64½ lb.

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*HINT.—To work this example by graphics, represent the weight by a vertical line drawn to scale; from one end of the line draw an indefinite line parallel to one of the segments of the rope, and from the other end of the line draw another indefinite line parallel to the other segment of the rope. These lines will intersect, and the distances from the point of intersection to the extremities of the vertical line will represent the tensions in the segments of the rope.

STRENGTH OF MATERIALS.

STRESSES AND STRAINS.

1926. When a force is applied to a body, it changes either its form or volume. A force, when considered with reference to the internal changes it tends to produce in any solid, is called a **stress.**

Thus, if a weight of 2 tons be held in suspension by a rod, the stress in the rod will be 2 tons. This stress is accompanied by a lengthening of the rod, which increases until the internal stress or resistance is in equilibrium with the external weight.

Stresses may be classified as follows:

Tensile, or pulling stress.

Compressive, or pushing stress.

Transverse, or bending stress.

Shearing, or cutting stress.

Torsional, or twisting stress.

1927. A **unit stress** is the amount of stress on a unit of area, and may be expressed either in pounds per square inch or in tons per square foot; or it is the load per square inch or square foot on any body.

Thus, if 10 tons are suspended by a wrought-iron bar which has an area of 5 square inches, the unit stress is 2 tons per square inch, because $\frac{10}{5} = 2$ tons.

1928. **Strain** is the deformation or change of shape of a body resulting from stress.

For example, if a rod 100 feet long is pulled in the direction of its length, and if it is lengthened 1 foot, it has a strain of $\frac{1}{100}$ of its length, or 1 per cent.

1929. **Elasticity** is the power which bodies have of returning to their original form after the external force on the body is withdrawn, providing the stress has not exceeded the elastic limit.

Consequently, we see from this that all material is

lengthened or shortened when subjected to either tensile or compressive stress, and the change of the length is directly proportional to the stress within the elastic limit.

For stresses within the elastic limits, materials are perfectly elastic, and return to their original length on removal of the stresses; but when their elastic limits are exceeded, the changes of their lengths are no longer regular, and a permanent **set** takes place; the destruction of the material has then begun.

1930. The **measure of elasticity** of any material is the change of length under stress within the elastic limit.

1931. The **elastic limit** is that unit stress under which the permanent set becomes visible.

The elasticity of wrought iron and that of steel are practically equal; that is, each material will change an equal amount of length under the same stress within the elastic limits.

The elastic limit of steel is higher than that of wrought iron; consequently, the former will lengthen or shorten more than the latter before its elasticity is injured.

TENSILE STRENGTH OF MATERIALS.

1932. The tensile strength of any material is the resistance offered by its fibers to being pulled apart.

The tensile strength of any material is proportional to the area of its cross-section.

Consequently, when it is required to find the safe tensile strength of any material, we have only to find the area at the minimum cross-section of the body, and multiply it by its strength per square inch, as given in Table 32 under the heading "Working Stress."

NOTE.—The minimum cross-section referred to in the above paragraph is that section of the material which is pierced with holes; such as bolt or rivet holes in iron, or knots in wood, if there are any.

1933. In Table 32 are given the average breaking and the working tensile stress of some materials.

The table shows that the tensile breaking strength

of cast iron is 16,000 pounds per square inch of cross-section, and that the working strength is from 1,500 to 3,500 pounds per square inch of cross-section.

TABLE 32.

Materials.	Breaking Stress in Pounds per Square Inch.	Working Stress in Pounds per Square Inch.
Timber................	10,000	600 to 1,200
Cast Iron..............	16,000	1,500 to 3,500
Wrought Iron..........	50,000	5,000 to 12,000
Steel.................	70,000	6,000 to 13,000

In machinery, such as steam-engines, where the parts are subjected to shocks, or are alternately compressed and extended, it is not safe to strain cast iron with more than 1,500 pounds per square inch of section, wrought iron with more than 5,000 pounds per square inch of section, or steel with more than 6,000 pounds per square inch of section.

But in structures in which the strains are constantly in one direction, as is the case with steam-boilers, wrought iron may be strained with from 6,000 to 8,000 pounds per square inch of section, or steel with from 8,000 to 10,000 pounds per square inch of section.

Consequently, strict attention must be given as to what working stress must be allowed for the materials of different structures.

NOTE.—For structures on which the load is applied suddenly, use the smaller working stresses given in the table, and for those on which the load is applied gradually, use the larger working stresses.

RULES AND FORMULAS FOR TENSILE STRENGTH.

1934. In the following formulas,

Let W = safe load in pounds;

    A = area of minimum cross-section;

    S = working stress in pounds per square inch, as given in Table 32.

Rule.—*The load in pounds on any bar subjected to a tensile strain is equal to the minimum sectional area of the bar, multiplied by the working stress in pounds per square inch, as given in Table 32.*

That is, $W = A S.$ **(118.)**

EXAMPLE.—A bar of good wrought iron which is 3 inches square is to be subjected to a steady tensile stress; what is the maximum load that it should carry ?

SOLUTION.—From what has been said above in regard to the materials and to the nature of the load, it will be safe in this case to use a working stress of 12,000 pounds per square inch.

Applying formula **118**, we have

$$W = 3 \times 3 \times 12,000 = 108,000 \text{ pounds.} \quad \textbf{Ans.}$$

1935. Rule.—*The minimum sectional area of any bar subjected to a tensile stress is equal to the load in pounds, divided by the working stress in pounds per square inch, as given in Table 32.*

That is, $A = \dfrac{W}{S}.$ **(119.)**

EXAMPLE.—What should be the area of a wrought-iron bar to carry a steady load of 66,000 pounds, if it is to resist a tensile stress of 12,000 pounds per square inch ?

SOLUTION.—Applying formula **119**,

$$A = \frac{66,000}{12,000} = 5.5 \text{ sq. in.} \quad \textbf{Ans.}$$

1936. Rule.—*The working stress in pounds per square inch is equal to the load in pounds divided by the minimum sectional area of the bar.*

That is, $S = \dfrac{W}{A}.$ **(120.)**

EXAMPLE.—A bar of wrought iron 3 inches square, subjected to tensile stress, carries a load of 86,400 pounds; what is the stress per square inch ?

SOLUTION.—Applying formula **120**,

$$S = \frac{86,400}{3 \times 3} = 9,600 \text{ lb. per sq. in.} \quad \textbf{Ans.}$$

STRENGTH OF CHAINS.

1937. Chains made of the same size iron vary in strength, owing to the different kinds of links from which they are made.

It is a good practice to anneal old chains which have become brittle by overstraining. This renders them less liable to snap from sudden jerks. It reduces their tensile strength, but increases their toughness and ductility, which are sometimes more important qualities.

1938. In the following formulas,

Let W = safe load in pounds sustained by chain;

D = diameter in inches of the iron from which the links are made.

Rule.—*The safe load in pounds of a stud-link wrought-iron chain is equal to 18,000, multiplied by the square of the diameter of the iron from which the links are made.*

That is, $W = 18,000\,D^2$.   **(121.)**

EXAMPLE.—What is the maximum load that should be carried by a stud-link wrought-iron chain, if its links are made from ¾-inch round iron?

SOLUTION.—Applying formula **121**, we have

$$W = 18,000 \times \tfrac{3}{4} \times \tfrac{3}{4} = 10,125 \text{ pounds. Ans.}$$

1939. Rule.—*The safe load in pounds of a close-link wrought-iron chain is equal to 12,000 multiplied by the square of the diameter of the iron from which the links are made.*

That is, $W = 12,000\,D^2$.   **(122.)**

EXAMPLE.—What is the maximum load that should be carried by a close-link wrought-iron chain, if its links are made from ¾-inch round iron?

SOLUTION.—Applying formula **122**, we have

$$W = 12,000 \times \tfrac{3}{4} \times \tfrac{3}{4} = 6,750 \text{ pounds. Ans.}$$

STRENGTH OF HEMP ROPES.

1940. The strength of hemp ropes does not depend so much upon the quality of the material and the cross-section of the rope as upon the method of manufacture and the amount of twisting.

The ropes in common use are three-strand shroud-laid rope, and hawser or cable-laid rope.

The strongest ropes are three-strand shroud-laid made without tar. Ropes made with tar are less flexible, and are reduced in strength about 25 per cent., but have better wearing qualities.

1941. In the following formulas,

Let W = maximum working load in pounds;

C = circumference of rope in inches.

Rule.—*The maximum working load in pounds that should be allowed on any hemp rope is equal to the square of the circumference of the rope multiplied by 100.*

That is, $W = 100\ C^2$.   **(123.)**

EXAMPLE.—What is the maximum load in pounds that should be carried by a hemp rope which has a circumference of 8 inches?

SOLUTION.—Substituting the value of C in formula **123**,

$$W = 100 \times 8^2 = 6,400\ \text{lb.}\quad \text{Ans.}$$

1942. **Rule.**—*The circumference of any hemp rope is equal to the square root of the maximum working load in pounds which it is capable of carrying, multiplied by .1.*

That is, $C = .1\sqrt{W}$.   **(124.)**

EXAMPLE.—A maximum working load of 1,000 pounds is to be carried by a hemp rope; what should be the circumference of the rope?

SOLUTION.—Applying formula **124**,

$$C = .1\sqrt{1,000} = 3.16\ \text{inches.}\quad \text{Ans.}$$

When measuring ropes, the circumference is sought instead of the diameter, because the ropes are not round and the circumference is not 3.1416 times the diameter. For three strands, the circumference is about 2.86 d; for seven strands, 3 d.

STRENGTH OF WIRE ROPES.

1943. Wire rope is made of iron and steel wire. It is stronger than hemp rope, and, to carry the same load, is of smaller diameter.

In substituting steel for iron rope, the object in view

should be to gain an increase of wear from the rope, rather than to reduce the size.

A steel rope to be serviceable should be of the best obtainable quality, because ropes made from low grades of steel are inferior to good iron ropes.

1944. In the following formulas,

Let W = maximum working load in pounds;

C = circumference of rope in inches.

Rule.—*The maximum working load in pounds that should be allowed on any wire rope is equal to the square of the circumference of the rope in inches, multiplied by 600.*

That is, $W = 600\, C^2$.   **(125.)**

Example.—What is the maximum load in pounds that should be carried by an iron wire rope whose circumference is 4½ inches?

Solution.—Applying formula **125,**

          • $W = 600 \times 4.5^2 = 12{,}150$ lb.   Ans.

1945. Rule.—*The circumference of any iron wire rope in inches is equal to the square root of the maximum working load in pounds multiplied by .0408.*

That is, $C = .0408\, \sqrt{W}$.   **(126.)**

Example.—A maximum working load of 12,150 pounds is to be carried by an iron wire rope; what should be the minimum circumference of the rope?

Solution.—Applying formula **126,**

          $C = .0408\, \sqrt{12{,}150} = 4\tfrac{1}{2}$ inches.   Ans.

1946. Rule.—*The above rules and formulas are also applicable when computing the safe strength of steel wire rope by substituting the constant 1,000 for the constant 600, and .0316 for .0408.*

Example.—What is the maximum load in pounds that should be carried by a steel wire rope, the circumference of which is 4½ inches?

Solution.—Applying formula **125** as modified by the rule,

          $W = 1{,}000 \times 4.5^2 = 20{,}250$ lb.   Ans.

Example.—A maximum working load of 10,485 pounds is to be

carried by a steel wire rope; what should be the minimum circumference of the rope?

SOLUTION.—Applying formula **126** as modified by the rule,

$$C = .0316\sqrt{10,485} = 3.24 \text{ inches. Ans.}$$

EXAMPLES FOR PRACTICE.

1. What should be the diameter of a steel piston-rod of a steam-engine to resist tension, if the piston is 19 inches in diameter and the pressure is 85 pounds per sq. in.? Ans. 2¼ in., nearly.

2. What safe load will a cast-iron bar of rectangular cross-section 7¼ inches by 3¼ inches support if subjected to shocks? The bar is in tension. Ans. 39,375 lb.

3. What is the stress per square inch on a piece of timber 8 inches square, which is subjected to a steady pull of 60,000 pounds? Ans. 937.5 lb. per sq. in.

4. What should be the safe load for a close-link wrought-iron chain whose links are made from ½-inch iron? Ans. 9,187.5 lb.

5. What safe load may a hemp rope carry whose circumference is 4 inches? Ans. 1,600 lb.

6. What should be the allowable working load for a steel wire rope whose circumference is 3¼ inches? Ans. 14,062.5 lb.

7. What should be the circumference of an iron wire rope to support a load of 20,000 pounds? Ans. 5¼ in., nearly.

CRUSHING STRENGTH OF MATERIALS.

1947. The crushing strength of any material is the resistance offered by its fibers to being pushed together.

If a bar is long compared with its cross dimensions, any slight disturbance from uniformity will cause it to bend sideways under the compressive force, and we have, then, not only compression, but compression compounded with bending.

To obtain only compression, the length of a rod should not be more than five times greater than its least diameter, or its least thickness when it is a rectangular rod.

Experimental tests on pillars have shown that their strengths are approximately inversely proportional to the squares of their lengths. That is, if there are two pillars of the same material, having the same cross-section, but

one is twice as long as the other, the long one will sustain only about one-quarter the load of the short one.

1948. Attention should be given to the ends of pillars, as their shape has great influence upon their strength. In Fig. 638 are shown three pillars with different shaped ends.

FIG. 638.

It has been proved by the aid of higher mathematics that, theoretically, a pillar having flat or fixed ends, as shown at *a*, is four times as strong as one that has round or movable ends, as shown at *c*, and one and seven-ninths times as strong as one having one flat and one round end, as shown at *b* ; *b* is thus two and one-fourth times as strong as *c*. It has also been found that if three pillars, *a*, *b*, *c*, which have the same cross-section, are to carry the same load and be of equal strength, their lengths must be as the numbers 2, 1½, and 1, respectively.
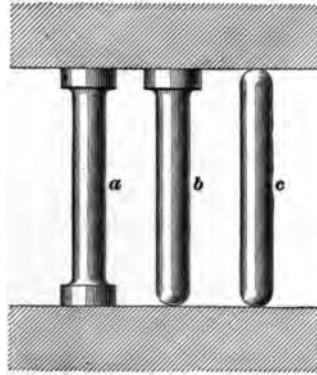
In practice, however, the ends of the pillars *b* and *c* are not generally made as shown by the figure, but have holes at their ends into which pins are fitted which are fastened to some other piece; as, for example, a connecting-rod of an engine. In such cases, it has been found that *a* is two times as strong as *c*, and that *b* is one and one-half times as strong as *c*. That is, in actual practice, a column fixed as at *c* is really one-half as strong as one fixed as at *a*, instead of being only one-quarter as strong, as given above.

Green or wet timber has only one-half the strength of dry and seasoned timber; consequently, its crushing strength is only one-half of that given in the table below.

1949. In Table 33 is given the mean crushing strength of some short specimens of materials in tons (of 2,000 pounds) per square inch.

24—7

TABLE 33.

Materials.	Crushing Strength in Tons per Square Inch.
Cast Iron................	40
Wrought Iron...........	18
Mild Steel...............	26
Cast Copper............	5
Cast Brass	4.5
Timber (Dry)	3.5
Brick...................	1
Stone...................	3

STRENGTH OF PILLARS.

1950. The following formula is applicable to pillars which are commonly used in practice, the lengths of which are about from 10 to 40 times their least diameter, or, if rectangular, their least thickness as indicated by d:

Let $C =$ crushing strength in tons per square inch of a short specimen of the material as given in Table 33;

$S =$ sectional area in square inches;

$L =$ length in inches;

$d =$ least thickness of rectangular pillar, or diameter of round pillar in inches;

$W =$ breaking load in tons;

$A =$ the area of the two flanges;

$B =$ the area of the web;

$a =$ a constant for the particular form of cross-section and material of which the pillar is made; its value is given in Tables 34 to 36 for such cross-sections as are given in the first column of those tables, and for such material as is mentioned at the top of the tables.

TABLE 34.

WROUGHT-IRON PILLARS.

Cross-section of Pillar.	When Both Ends of the Pillar are Flat or Fixed.	When One End of the Pillar is Flat or Fixed, and the Other Round or Movable.	When Both Ends of the Pillar are Round or Movable.
Round.	2,250	1,500	1,125
Square or Rectangle.	3,000	2,000	1,500
Thin Square Tube.	6,000	4,000	3,000
Thin Round Tube.	4,500	3,000	2,250
Angle with Equal Sides.	1,500	1,000	750
Cross with Equal Arms.	1,500	1,000	750
I Beam.	$3,000 \times \dfrac{A}{A+B}$	$2,000 \times \dfrac{A}{A+B}$	$1,500 \times \dfrac{A}{A+B}$

1951. Rule.—*The breaking load of a pillar in tons is equal to the crushing strength of a short specimen of the material as given in Table 33, multiplied by the sectional area of the pillar in square inches, and the product divided by*

TABLE 35.
CAST-IRON PILLARS.

Cross-section of Pillar.	When Both Ends of the Pillar are Flat or Fixed.	When One End of the Pillar is Flat or Fixed, and the Other Round or Movable.	When Both Ends of the Pillar are Round or Movable.
Round.	281.25	187.5	140.625
Square or Rectangle.	375.00	250.0	187.500
Thin Square Tube.	750.00	500.0	375.000
Thin Round Tube.	562.50	375.0	281.250
Angle with Equal Sides.	187.50	125.0	93.750
Cross with Equal Arms.	187.50	125.0	93.750
I Beam.	$375 \times \dfrac{A}{A+B}$	$250 \times \dfrac{A}{A+B}$	$125 \times \dfrac{A}{A+B}$

the result obtained by dividing the square of the length of the pillar in inches by the square of the diameter, or least thickness if rectangular, multiplied by the value of a, plus 1.

That is, $W = \dfrac{C\,S}{\dfrac{L^2}{a\,d^2}+1}.$ **(127.)**

TABLE 36.

WOODEN PILLARS.

Cross-section of Pillar.	When Both Ends of the Pillar are Flat or Fixed.	When One End of the Pillar is Flat or Fixed, and the Other Round or Movable.	When Both Ends of the Pillar are Round or Movable.
Round.	187.5	125.00	93.75
Square or Rect-angle.	250.0	166.66	125.00
Hollow Square Made of Boards.	500.0	333.33	250.00

The result obtained by the formula must be divided by 6 to get the safe working load.

NOTE.—If the length of the pillar is given in feet, be sure to reduce it to inches before substituting in the formula.

EXAMPLE.—A wooden pillar 6 inches square and 144 inches long is fixed at both ends; what load will it sustain with safety?

SOLUTION.—Using formula **127,** we have

$$W = \frac{3.5 \times 6 \times 6}{\frac{144 \times 144}{250 \times 6 \times 6} + 1} = 38.14 \text{ tons, nearly.}$$

Which, divided by 6, gives $\frac{38.14}{6} = 6.357$ tons, or the load it is capable of sustaining with safety. Ans.

EXAMPLE.—A wrought-iron pillar 4 inches in diameter and 60 inches long is fixed at one end and movable at the other; what load will it sustain with safety?

SOLUTION.—Using formula **127,**

$$W = \frac{18 \times 4 \times 4 \times .7854}{\frac{60 \times 60}{1,500 \times 4 \times 4} + 1} = 196.69 \text{ tons.}$$

Which, divided by 6, gives $\dfrac{196.69}{6} = 32.78$ tons, nearly, or the load it is capable of sustaining with safety.   Ans.

EXAMPLE.—A cast-iron pillar is 20 feet long and its cross-section is a cross with equal arms which are 1 inch thick and 10 inches long. (See dimension d, Table 35.)   The two ends of the pillar are movable. What load will the column safely sustain ?

SOLUTION.—Area of cross-section is equal to $(10 \times 1) + 2(4.5 \times 1) = 19$ square inches; 20 feet are equal to 240 inches.
Now, applying formula **127**,

$$W = \dfrac{40 \times 19}{\dfrac{240 \times 240}{93.75 \times 10 \times 10} + 1} = 106.88 \text{ tons.}$$

Which, divided by 6, gives $\dfrac{106.88}{6} = 17.73$ tons, the load it is capable of sustaining with safety.   Ans.

When using formula **127**, first obtain the value of C from Table 33.   Next, calculate the area of the cross-section of the pillar.   Then, find the value of a from one of the tables.   Finally, be sure that the length of the pillar has been reduced to inches before substituting in the formula.

_____

EXAMPLES FOR PRACTICE.

1.   What load may be safely carried by a hollow cylindrical cast-iron pillar 20 ft. long, inside diameter 8″, and outside diameter 10″? Both ends of the pillar are fixed.   Ans. 93.13 tons.

2.   A rectangular wooden column is 14 ft. long, and has one end rounded; if the cross-section is 12″ × 8″, what load will be required to break it ?   Ans. 92.15 tons.

3.   A solid wrought-iron column, which has both ends movable, is 8″ in diameter and 8 ft. long; what load will it safely support ?
                    Ans. 11.1 tons.

_____

TRANSVERSE STRENGTH OF MATERIALS.

1952.   The transverse strength of any material is the resistance offered by its fibers to being broken by bending. As, for example, when a beam, bar, rod, etc., which is supported at its ends, is broken by a force applied between its supports.

The transverse strength of any beam, bar, rod, etc., is proportional to the product of the square of its depth multiplied by its width; consequently, it is more economical to increase the depth than the width.

TABLE 37.
CONSTANTS FOR TRANSVERSE STRENGTH.

Materials.	Safe Transverse Strength in Pounds.	Materials.	Safe Transverse Strength in Pounds.
METALS.		WOODS.	
Cast Iron........	100	Birch...........	35
Wrought Iron....	150	Elm	25
Structural Steel..	160	Ash	45
Copper..........	50	Beech..........	30
Brass	55	Hickory	50
		Maple	60
		Oak (American)..	45
		Pine (Pitch)	40
		Pine (White)....	30

1953. A **cantilever** is a beam, bar, rod, etc., fixed at one end and subjected to a transverse stress, as shown in Fig. 639. It has a tendency to overthrow the wall or structure to which it is attached.

The strength of a cantilever varies inversely as the distance of the load from the section acted upon; and the stress upon any section varies
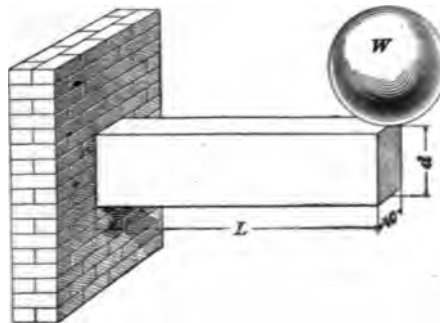
FIG. 639.

directly as the distance of the load from that section

The strength of a beam, bar, rod, etc., which has both its ends supported, but not fixed, and which is loaded in the middle between its supports, is four times greater than when it is fixed at one end only.

A cantilever uniformly loaded will sustain twice as great a load as one on which the load is applied at one end; and a beam resting on two supports uniformly loaded will sustain twice as great a load as one on which the load is applied in the middle, between its supports.

In Table 37 is given the safe transverse strength of bars of different kinds of material, 1 inch square and 1 foot long, with the load suspended from one end.

RULES AND FORMULAS FOR THE TRANSVERSE STRENGTH OF BEAMS.

1954. In the following formulas,

Let d = the depth of beam in inches;

d_1 = diameter of cylindrical beam in inches;

w = the width of the beam in inches;

L = the length of the beam in feet between its supports;

S = the safe transverse strength as given in the above table;

W = the safe load in pounds.

For a rectangular or square cantilever to which the load is applied at one end, as shown in Fig. 639:

Rule.—*To find the maximum safe load in pounds that should be allowed at the end of any rectangular or square cantilever, multiply the square of the depth in inches by the width in inches and by the safe transverse strength of the material as given in Table 37; divide this product by the length in feet.*

That is, $W = \dfrac{d^2 w S}{L}$.   **(128.)**

EXAMPLE.—What is the maximum safe load that can be placed at one end of a cast-iron bar which projects 4 feet, the depth being 6 inches and the width 3 inches?

SOLUTION.—Applying formula **128,** we have

$$W = \frac{6 \times 6 \times 3 \times 100}{4} = 2,700 \text{ pounds. Ans.}$$

1955. For a cylindrical cantilever to which the load is applied at one end, as shown in Fig. 640:

Rule.—*To find the maximum safe load in pounds that should be allowed at the end of any cylindrical cantilever, multiply the cube of its diameter in inches by .6 of the safe transverse strength of the material as given in Table 37, and divide the product by the length in feet.*
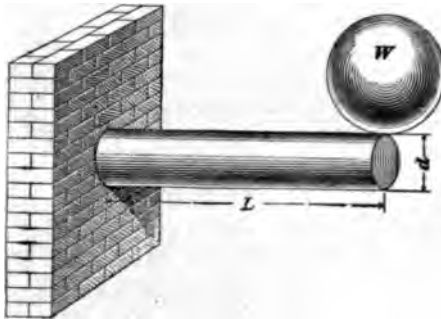
FIG. 640.

That is, $W = \dfrac{d_1^3 \times .6\,S}{L}.$ **(129.)**

EXAMPLE.—What is the maximum load that can be placed with safety at one end of a cast-iron bar 4 inches in diameter that projects 3 feet?

SOLUTION.—Applying formula **129,** we have

$$W = \frac{4 \times 4 \times 4 \times .6 \times 100}{3} = 1,280 \text{ pounds. Ans.}$$

1956. When the load is uniformly distributed on a cantilever of any cross-section, as shown in Fig. 641, it will sustain a load twice as great as when the load is applied at one end. For example, if the cantilevers in the two examples above were to carry a uniformly distributed load, they would sustain $2,700 \times 2 = 5,400$

FIG. 641.

pounds and $1,280 \times 2 = 2,560$ pounds, respectively.
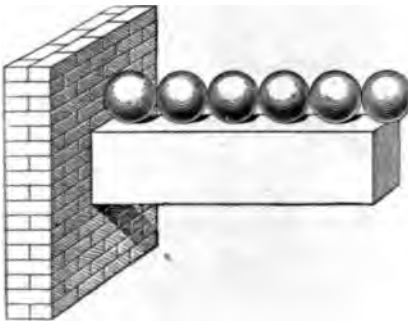
1957. For a rectangular or square beam the ends of which merely rest upon supports, and loaded in the middle, as shown in Fig. 642:
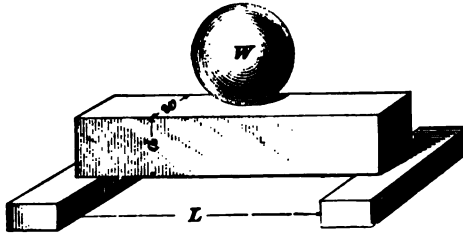
Rule.—*To find the maximum safe load in pounds that any rectangular or square beam is capable of sustaining at the middle when its ends merely rest upon supports,*

FIG. 642.

multiply four times the square of its depth in inches, by its width in inches, and by the safe strength of the material as given in Table 37 ; divide this product by the distance between its supports in feet ;

or,
$$W = \frac{4\,d^2\,w\,S}{L}. \qquad \textbf{(130.)}$$

EXAMPLE.—What maximum safe load is a bar of cast iron capable of sustaining in the middle between its supports on which its ends merely rest, if its depth is 6 inches, its width 3 inches, and the distance between the supports is 4 feet ?

SOLUTION.—Applying formula **130**,

$$W = \frac{4 \times 6^2 \times 3 \times 100}{4} = 10{,}800 \text{ lb.} \quad \text{Ans.}$$

1958. For a cylindrical beam supported at its ends and loaded in the middle, as shown in Fig. 643:

Rule.—*To find the maximum safe load in pounds that any cylindrical beam is capable of sustaining at the middle when its*
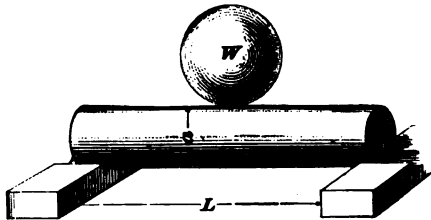
FIG. 643.

ends merely rest upon supports, multiply four times the cube of its diameter by .6 of the safe strength of the material as given in Table 37 ; divide this product by the distance between its supports in feet ;

or,
$$W = \frac{4\,d_1^2 \times .6\,S}{L}. \qquad \textbf{(131.)}$$

EXAMPLE.—What maximum safe load is a bar of cast iron capable of sustaining in the middle between its supports, on which its ends merely rest, if it is 4 inches in diameter, and if the distance between its supports is 3 feet ?

SOLUTION.—Applying formula **131**,

$$W = \frac{4 \times 4^3 \times .6 \times 100}{3} = 5,120 \text{ lb.}   \text{Ans.}$$

1959. When the load is uniformly distributed on a beam of any cross-section, as shown in Fig. 644, it will sustain a load twice as great as when the load is applied in the middle between the supports.
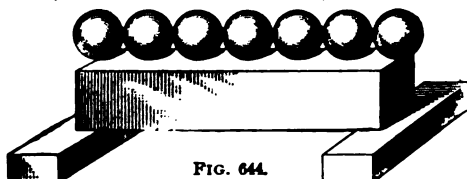
FIG. 644.

For example, if the beams in the last two examples were to carry a uniformly distributed load, they would sustain $10,800 \times 2 = 21,600$ pounds, and $5,120 \times 2 = 10,240$ pounds, respectively.

SHEARING OR CUTTING STRENGTH OF MATERIALS.

1960. The shearing strength of any material is the resistance offered by its fibers to being cut in two.

Thus the pressure of the cutting edges of an ordinary shearing machine, Fig. 645, causes a shearing stress in the plane $a\,b$. The unit shearing force may be found by dividing the force P by the
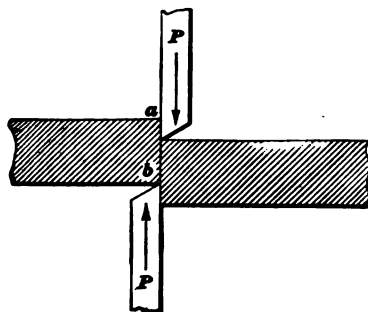
FIG. 645.

area of the plane $a\,b$.

1961. Fig. 646 shows a piece in double shear; here the central piece $c\,d$
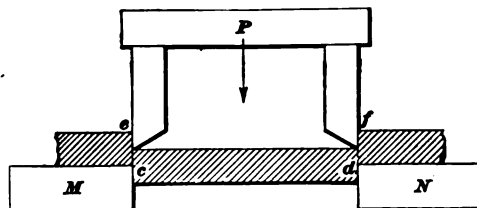
FIG. 646.

is forced out while the ends remain on their supports M and N.

The shearing strength of any body is directly proportional to its area.

1962. In Table 38 are given the greatest and the safe shearing strengths per square inch of different kinds of materials:

TABLE 38.

Materials.	Greatest Shearing Stress in Pounds per Square Inch.	Safe Shearing Stress in Pounds per Square Inch.
Cast Iron........	18,000	1,500 to 3,000
Wrought Iron.....	40,000	4,000 to 10,000
Steel.......	60,000	5,000 to 12,000

1963. In the following formula,

Let a = area of cross-section in inches;

      S = safe shearing stress, as given in Table 38;

      W = safe load in pounds.

Rule.—*The safe load that any body which is subjected to a shearing stress is capable of sustaining is equal to the area of its cross-section in inches, multiplied by its safe shearing stress, as given in Table 38.*

That is, $W = a\,S.$ **(132.)**

EXAMPLE.—If the beam in Fig. 646 is made of wrought iron 4 inches in depth and 2 inches in width, what steady shearing stress is it capable of sustaining with safety?

SOLUTION.—Applying formula **132**, $W = 4 \times 2 \times 10,000 = 80,000$ lb. This result must be multiplied by 2, since the beam is sheared in two places, along the lines $c\,c$ and $f\,d$. Hence, the stress which the beam will safely sustain is $80,000 \times 2 = 160,000$ lb.   Ans.

EXAMPLE.—What force is required to punch a hole $\frac{3}{4}''$ in diameter through a steel plate $\frac{1}{4}''$ thick?

SOLUTION.—It is evident that punching is but shearing in a circle instead of a straight line. The area punched (sheared) is equal to the

thickness of the plate multiplied by the circumference of a circle having the same diameter as the punched hole. For, if the plate were cut through one of the diameters of the punched hole and the two semicircles were straightened out, the punched surface would be a rectangle, which would have a length equal to the circumference of a circle whose diameter was equal to that of the hole, and a breadth equal to the thickness of the plate. In this case, the area $= \frac{1}{4} \times$ 3.1416 $\times \frac{1}{4} = .98175$ sq. in. Table 38 gives the ultimate shearing strength · of steel as 60,000 lb. per sq. in. Hence, the total force required is .98175 $\times$ 60,000 $=$ 58,905 lb. Ans.

EXAMPLES FOR PRACTICE.

1. What is the greatest load that can be safely carried by a steel rectangular cantilever at its extreme end, if the bar is 2" wide, 3" deep, and 2 ft. 6" long ?   Ans. 1,152 lb.

2. What is the greatest uniform load that can be safely carried by a white pine girder 6" wide, 8" deep, 16 ft. long, and supported at its ends ?   .   Ans. 5,760 lb.

3. A cast-iron bar 1¼" in diameter and 5 ft. 3" long is supported at its ends; what load will it safely sustain in the middle ?   Ans. 245 lb.

4. What force is required to punch a 1¼" hole through a wrought-iron plate $\frac{7}{16}$" thick ?   Ans. 68,723 lb.

5. What force is required to cut off the end of a cast-iron bar whose diameter is 2¼" ?   Ans. 88,357 lb.

LINE SHAFTING.

1964. A line of shafting is one continuous run, or length, composed of lengths of shafts joined together by couplings.

The **main line** of shafting is that which receives the power from the engine or motor, and distributes it to the other lines of shafting, or to the various machines to be driven.

Line shafting is supported by hangers, which are brackets provided with bearings bolted either to the walls, posts, ceilings, or floors of the building. Short lengths of shafting called **countershafts** are provided to effect changes of speed, and to enable the machinery to be stopped or started.

Shafting is usually made cylindrically true, either by a special rolling process known as **cold-rolled shafting,** or

else it is turned up in a machine called a lathe. In the latter case, it is called **bright shafting.** What is known as **black shafting** is simply bar iron rolled by the ordinary process, and turned where it receives the couplings, pulleys, bearings, etc.

Bright-turned shafting varies in diameter by $\frac{1}{4}$ of an inch to about $3\frac{1}{2}$ inches in diameter; above this diameter the shafting varies by $\frac{1}{2}$ inch. The actual diameter of a bright shaft is $\frac{1}{16}$ of an inch less than the actual commercial diameter, it being designated from the diameter of the ordinary round bar-iron from which it is turned. Thus, a length of what is called 3-inch bright shafting is only $2\frac{15}{16}$ inches in diameter.

Cold-rolled shafting is designated by its actual commercial diameter; thus, a length of what is called 3-inch shafting is 3 inches in diameter.

1965. In Table 39 is given the maximum distance between the bearings of some continuous shafts which are used for the transmission of power.

TABLE 39.

Diameter of Shaft in Inches.	Distance Between Bearings in Feet.	
	Wrought-Iron Shaft.	Steel Shaft.
2	11	11.5
3	13	13.75
4	15	15.75
5	17	18.25
6	19	20.00
7	21	22.25
8	23	24.00
9	25	26.00

The necessary diameters of the various lengths of shafts composing a line of shafting should be proportional to the

quantity of power delivered by each respective length.   In this connection, the positions of the various pulleys depend upon the distance between the pulley and the bearing and upon the amount of power given off by the pulleys.   Suppose, for example, that a piece of shafting delivers a certain amount of power; then, it is obvious that the shaft will deflect or bend less if the pulley transmitting that power be placed close to a hanger or bearing than if it be placed midway between the two hangers or bearings.

Note.—It is impossible to give any rule for the proper distance of bearings which could be used universally, as in some cases the requirements demand that the bearings be nearer together than in others.

1966.   To compute the horsepower that can be transmitted by a shaft of any given diameter:

Let D = diameter of shaft;

R = revolutions per minute;

H = horsepower transmitted;

C = constant taken from the following table:

TABLE 40.
CONSTANTS FOR LINE SHAFTING.

Material of Shaft.	No Pulleys Between Bearings.	Pulleys Between Bearings.
Steel or Cold-Rolled Iron ..	65	85
Wrought Iron	70	95
Cast Iron	90	120

Rule.—*The horsepower that a shaft will transmit is equal to the product of the cube of the diameter and the number of revolutions, divided by the value of C for the given material.*

That is, $H = \dfrac{D^3 R}{C}$.   **(133.)**

Example.—What horsepower will a 3-inch wrought-iron shaft transmit, which makes 100 revolutions per minute, and has pulleys between bearings?

Solution.—Applying formula **133**, we have

$$H = \frac{3 \times 3 \times 3 \times 100}{95} = 28.42 \text{ horsepower.   Ans.}$$

1967. To compute the number of revolutions a shaft must make to transmit a given horsepower:

Rule.—*The number of revolutions necessary for a given horsepower is equal to the product of the value of the constant C for the given material and the number of horsepower, divided by the cube of the diameter.*

That is, $R = \dfrac{CH}{D^3}$. **(134.)**

EXAMPLE.—How many revolutions must a 2-inch steel shaft make per minute to transmit 16 horsepower? There are no pulleys between bearings.

SOLUTION.—Applying formula **134**, we have $\dfrac{65 \times 16}{2 \times 2 \times 2} = 130$ revolutions. Ans. .

1968. To compute the diameter of a shaft that will transmit a given horsepower, the number of revolutions the shaft makes per minute being given:

Rule.—*The diameter of a shaft equals the cube root of the quotient obtained by dividing the product of the value of the constant C for the given material and the number of horsepower by the number of revolutions.*

That is, $D = \sqrt[3]{\dfrac{CH}{R}}$. **(135.)**

EXAMPLE.—What must be the diameter of a cast-iron shaft to transmit 22.5 horsepower? The shaft makes 100 revolutions per minute, and there are pulleys between bearings.

SOLUTION.—Applying formula **135**, we have

$$D = \sqrt[3]{\dfrac{120 \times 22.5}{100}} = 3 \text{ in. Ans.}$$

1969. As the speed of shafting is used as a multiplier in the calculations of the horsepower of shafts, a shaft having a given diameter will transmit more power in proportion as its speed is increased. Thus, a shaft which is capable of transmitting 10 horsepower when making 100 revolutions per minute will transmit 20 horsepower when making 200 revolutions per minute. We may, therefore,

say *the horsepowers transmitted by two shafts are directly proportional to the number of revolutions.*

EXAMPLES FOR PRACTICE.

1. What horsepower will a $2\frac{1}{4}''$ wrought-iron shaft transmit when running at 110 revolutions per minute, it being used for transmission only ?   Ans. 24.55 horsepower.

2. A $6''$ cast-iron shaft transmits 150 horsepower; how many revolutions per minute must it make, there being no pulleys between bearings ?   Ans. 62.5 R. P. M.

3. What should be the diameter of a wrought-iron shaft to transmit 100 horsepower at 150 revolutions per minute, there being pulleys between bearings ?   Ans. 6.82 in.

4. A certain line shaft is to transmit to a number of machines by means of pulleys between its bearings 65 horsepower when running at 150 revolutions per minute; what should be its diameter ?

                          Ans. $3\frac{1}{4}$ in., nearly
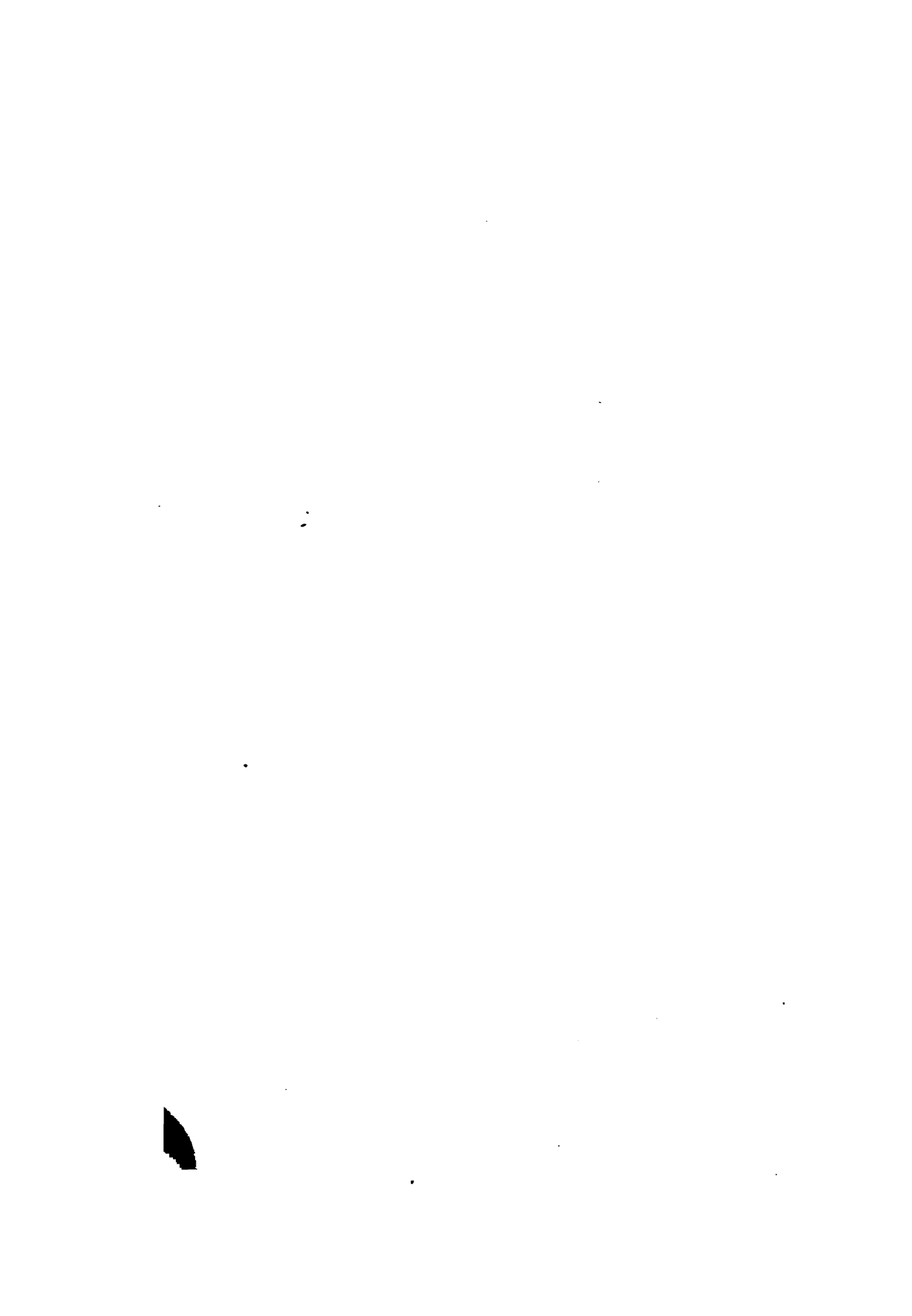
STEAM AND STEAM-BOILERS.

HEAT.

NATURE OF HEAT.

1970. **Heat** *is a form of energy.* It is the effect of a motion of the molecules composing matter. It has been stated in Mechanics that all matter is composed of molecules; now, these molecules are not in a state of rest, but are moving or vibrating back and forth with a greater or less velocity, and it is this movement of the molecules that causes the sensations of warmth or cold. If the motion is slow, the body appears cold to the touch; when the vibrations are rapid, the body becomes warm or hot.

1971. It was shown in Mechanics that a body in motion has kinetic energy, the amount of which is measured in foot-pounds by multiplying the weight of the body by the square of its velocity and dividing by 64.32. Since the molecules composing matter are in motion, they must possess kinetic energy, and we are justified, therefore, in saying that *heat is a form of energy.*

1972. **Temperature** is a term used to indicate how hot or cold a body is; i. e., to indicate the rate of vibration of the molecules of a body. A hot body has a high temperature; a cold body, a low temperature. When a body, as, for example, an iron bar, receives heat from any source, its temperature rises; on the other hand, when a body loses heat, its temperature falls.

1973. The temperature is *not* a measure of the quantity of heat a body possesses. *Temperature* may be considered to be a measure of the velocity of the molecules of a body as they vibrate to and fro, while the *quantity of heat* may

§ 18

be considered to be the kinetic energy of the molecules composing the body.   A small iron rod may be heated to whiteness and yet possess a very small quantity of heat.   Its temperature is very high, which simply indicates that the molecules of the rod are vibrating with an extremely high velocity.

1974.   Temperature is measured by an instrument called the thermometer, which is so familiar as to scarcely need description.   It consists of a thin glass tube, at one end of which is a bulb filled with mercury.   Upon being heated, the mercury expands in proportion to its temperature.   Thermometers are graduated in different ways.   In the Fahrenheit thermometer, which is generally used in this country, the point where the mercury stands when the instrument is placed in melting ice is marked 32°.   The point indicated by the mercury when the thermometer is placed in water boiling in open air at the level of the sea is marked 212°.   The tube between these two points is divided into 180 equal parts called degrees.

1975.   Effects of Heat.—Suppose we take a vessel filled with some substance, say water.   Let the vessel be a cylinder fitted with a piston, as shown in Fig. 647.   The water is, say, at the freezing-point, and the millions of molecules composing the water are moving to and fro with a comparatively small velocity.   Suppose the vessel is placed in a fire or furnace.   Heat is communicated to the molecules of water, and they begin to move faster and faster. That is, their kinetic energy increases, and if a thermometer were inserted in the vessel it would be found that the temperature of the water rises.   Consequently, one effect of heat is to raise the temperature of the body to which it is applied.   But, after reaching a certain temperature, the molecules of the water not only move faster, but they move

FIG. 647.

farther from each other, and their paths are longer. It is plain that if the molecules are farther apart than they were originally, the whole body of them must take up more space. In other words, after reaching a certain temperature, the water expands as heat is added. Hence, another effect of heat is to expand bodies to which it is applied. Common examples of the expansion of bodies by heat are seen in the setting of tires, the expansion of the rails of a railway in summer, etc.

1976. The heat supplied to the vessel of water has so far done three things: (1) Raised the temperature of the water, and thus increased the kinetic energy of the molecules. Let the amount of heat expended for this purpose be denoted by S. (2) A certain quantity of heat has been used in expanding the water; that is, in pushing its molecules farther apart against the force of cohesion. Denote the amount of heat so expended by I. (3) Since the water expands, it must raise the piston P against the pressure of the atmosphere, and, consequently, more heat must be used to expand the water than would be required if there were no pressure on the upper side of the piston. Call this extra quantity of heat W.

If we denote by Q the total heat given up to the vessel of water, we have

$$Q = S + I + W.$$

Ordinarily, the greater part of the heat given to a body is spent in raising its temperature, and but little is used in expanding the body. That is, the quantity S is nearly equal to the quantity Q, while the quantities I and W are nearly nothing.

1977. Suppose that the piston is removed from the cylinder of Fig. 647, and a thermometer is inserted. As the vessel becomes more and more heated, the temperature indicated by the thermometer will rise until it reaches $212°$. So far most of the heat has been used to raise the temperature of the water. But now, no matter how much heat is added to the water, the thermometer stands at $212°$, and

can not be made to rise higher. This is the reason: When the temperature reaches 212°, the molecules of water have been set into such rapid motion that the force of cohesion is no longer able to hold them, and they tend to separate. In other words, the water is changing to a gas (steam), and all of the heat is being used to effect this change. The temperature of the steam will remain at 212° until all the water is changed to steam; then, if more heat is applied, the temperature of the steam will begin to rise.

Suppose we take a block of ice at a temperature of say 14°, and heat it. If a thermometer is placed in contact with the ice, its temperature will rise until it reaches 32°, and will then remain stationary. As soon as this temperature is reached, the ice begins to melt or change to water, and the heat, instead of raising the temperature farther, is all used to effect this change of state. Here, then, is another effect produced by heat. It will change a solid to a liquid, or a liquid to a gas.

1978. Latent Heat.—The heat which is expended in changing a body from the solid to the liquid state, or from the liquid to the gaseous state, is called *latent heat.* The portion of the heat applied which raises temperature, and which, therefore, affects the thermometer, is sometimes called **sensible heat.**

1979. Measurement of Heat.—Since heat is not a substance, it can not be measured directly in pounds or quarts; but, like force, it may be measured by the effects it produces. Suppose a certain quantity of heat raises the temperature of a pound of water from 52° to 53°, it will take the same quantity of heat to raise the temperature of a pound from 53° to 54°, and it will take double the quantity to raise the temperature of the pound of water from 52° to 54° that it took to raise the temperature from 52° to 53°. The unit quantity of heat is the quantity required to raise the temperature of a pound of water from 62° to 63°. This unit is called the **British thermal unit,** or B. T. U.

1980. Relation Between Heat and Work.—Suppose that, in the experiment shown in Fig. 647, the piston had been allowed to remain in the cylinder while the water was being changed to steam. Steam at 212° occupies nearly 1,700 times the space that the water originally occupied. Hence, the piston would be lifted in the cylinder to give room for the steam which was being formed. But to raise the piston requires work. Here, then, is an example of *work being performed by heat.* On the other hand, work will produce heat. If two blocks of wood are rubbed briskly together, they will become warm, and may even ignite. The work of friction causes the journals and bearings of fast-running machines to heat. A small iron rod may be heated to redness by pounding it on an anvil.

1981. Since work may be changed into heat, and heat into work, it seems probable that there is some fixed ratio between the unit of heat (B. T. U.) and the unit of work, the foot-pound. By a careful series of experiments, Dr. Joule, of England, discovered this ratio. He found that one B. T. U. is equal to 772 foot-pounds; later and more careful experiments show that 778 foot-pounds is more nearly correct. This number, 778 foot-pounds, is called the **mechanical equivalent** of one B. T. U.

We have, then, the following important law: *Heat may be changed to work, or work to heat ; 778 foot-pounds of work are required to produce 1 B. T. U., and, conversely, the expenditure of 1 B. T. U. produces 778 foot-pounds of work.*

EXAMPLE 1.—The burning of a pound of coal gives out sufficient heat to raise 14,000 pounds of water from 62° to 63°. If all this heat is wholly utilized, how high will it lift a weight of 700 pounds ?

SOLUTION.—Since 1 B. T. U. raises a pound of water from 62° to 63°, it requires 14,000 B. T. U. to raise 14,000 lb. of water from 62° to 63°. Hence, the burning of the pound of coal gives out 14,000 B. T. U. One B. T. U. is equivalent to 778 foot-pounds; hence, 14,000 B. T. U. are equivalent to $14,000 \times 778 = 10,892,000$ foot-pounds. Then, the height to which the weight can be raised is $10,892,000 \div 700 = 15,560$ feet. Ans.

EXAMPLE 2.—A cannon-ball weighing 60 pounds moves with a

velocity of 1,300 ft. per sec. Suppose the ball were suddenly stopped and its kinetic energy changed into heat. How many B. T. U. would be developed? If all this heat were applied to 100 pounds of water at a temperature of 60°, to what temperature would the water be raised?

SOLUTION.—By formula **113,** *Mechanics,* Part 1, the kinetic energy of the cannon-ball is $\frac{W v^2}{64.32} = \frac{60 \times 1,300^2}{64.32} = 1,576,492$ foot-pounds. But 778 foot-pounds = 1 B. T. U. Therefore, the number of B. T. U. developed is $1,576,492 \div 778 = 2,026.3$ B. T. U. Since 1 B. T. U. raises the temperature of a pound of water 1 degree, it will take 100 B. T. U. to raise 100 pounds of water 1 degree. Hence, 2,026.3 B. T. U. will raise 100 pounds of water $2,026.3 \div 100 = 20.263°$, and the final temperature of the water will be $60° + 20.263° = 80.263°$. Ans.

1982. Specific Heat.—One B. T. U. raises the temperature of one pound of water one degree; will it have the same effect on a pound of mercury? Heat two one-pound iron balls to the temperature of boiling water, 212°; having now the same weights and temperatures, each ball has the same quantity of heat. Place one of these balls in a vessel, into which pour slowly enough water at a temperature of 60° that the iron will be cooled to 70° while the water is heated to the same temperature. Now, place the other hot ball in another vessel, into which pour mercury at a temperature of 60° until the iron and mercury reach a common temperature of 70°. In each case the hot ball was cooled from 212° to 70°; each, therefore, gave up the same quantity of heat. When, however, we consider its effects, we find that it raised less than $\frac{1}{2}$ pound of water through a range of 10°, while $14\frac{1}{2}$ pounds of mercury, nearly 30 times as much, was raised through the same range. It is plain, therefore, that to raise a pound of mercury from 62° to 63° requires $\frac{1}{30}$ the heat necessary to raise a pound of water from 62° to 63°. Hence, we say the *specific heat* of the mercury is $\frac{1}{30}$, or .0333.

The **specific heat** *of a body is the ratio between the quantity of heat required to warm that body one degree and the quantity of heat required to warm an equal weight of water one degree.*

EXAMPLE 1.—It is found that to raise the temperature of 20 pounds of iron from 62° to 63° requires 2.276 B. T. U. What is the specific heat of iron ?

SOLUTION.—To raise 20 pounds of water from 62° to 63° requires 20 B. T. U. The specific heat of the iron is, according to the above definition, the ratio between the quantities of heat required to warm the iron and the water, respectively, through 1 degree; that is, it is the ratio 2.276 : 20 = 2.276 ÷ 20 = .1138. Ans.

EXAMPLE 2.—The specific heat of silver is .057. How many B. T. U. are required to raise 22 pounds of silver from 50° to 60° ?

SOLUTION.—To raise the temperature of a pound of water 1 degree requires 1 B. T. U. Since the specific heat of silver is .057, only .057 B. T. U. is required to raise 1 pound of silver 1 degree. Hence, to raise 22 pounds of silver 10 degrees must require .057 × 22 × 10 = 12.54 B. T. U. Ans.

1983.   Rule.—*To find the number of B. T. U. required to raise the temperature of a body a given number of degrees, multiply the specific heat of the body by its weight in pounds and by the number of degrees.*

Denote the number of B. T. U. required by U; the specific heat by c; the weight by W, and let t and t_1 be the temperatures before and after the heat is applied, respectively.

Then, $U = c\, W\, (t_1 - t)$.   **(138.)**

The specific heats of some of the more common substances are given in the following table:

TABLE 41.

Substance.	Sp. Heat.	Substance.	Sp. Heat.
Water.............	1.0000	Ice.................	.5040
Sulphur	.2026	Steam (superheated).	.4805
Iron..............	.1138	Air	.2375
Copper	.0951	Oxygen	.2175
Silver.............	.0570	Hydrogen	3.4090
Tin...............	.0562	Nitrogen	.2438
Mercury	.0333	Carbon monoxide....	.2479
Lead	.0314	Carbon dioxide	.2170

1984. Latent Heat of Fusion.—This term is applied to the quantity of heat required to change a pound of a given substance from the solid to the liquid state. The only case of interest to the engineer is the heat required to change a pound of ice to water. Careful experiments show that about 144 B. T. U. are required to change a pound of ice at 32° to water at 32°. Hence, the latent heat of water is 144 B. T. U.

1985. The **latent heat of steam** is the quantity of heat required to change a pound of water at 212° into steam at 212°. Experiment shows that this quantity of heat is about 966 B. T. U. This shows that the heat required to change a pound of water at 212° to steam is 966 times as great as the quantity required to raise the temperature of a pound of water from 62° to 63°. The latent heat of steam is different for different temperatures.

EXAMPLE.—How many B. T. U. are required to change 5 pounds of ice at 15' into steam at 212° ?

SOLUTION.—The heat units required to raise the temperature of the ice from 15' to 32° (the melting temperature) is, according to formula **136,**

$$U = c\ W(t_1 - t) = .504 \times 5 \times (32 - 15) = 42.84\ \text{B. T. U.}$$

To change the ice to water requires 144 B. T. U. for each pound, or 144 × 5 = 720 B. T. U. To raise the water from 32° to 212° requires, according to formula **136,**

$$c\ W(t_1 - t) = 1 \times 5 \times (212 - 32) = 5 \times 180 = 900\ \text{B. T. U.}$$

Finally, to change the water to steam requires 966 B. T. U. per pound, or 966 × 5 = 4,830 B. T. U. Therefore, in all, 42.84 + 720 + 900 + 4,830 = 6,492.84 B. T. U. are required. Ans.

Expressed in foot-pounds, the work required to effect the above change would be 6,492.84 × 778 = 5,051,429.5 foot-pounds, or work enough to lift a weight of 1,000 pounds nearly a mile.

1986. Since a pound of ice requires 144 B. T. U. to change it to water, it follows that when a pound of water at 32° changes to ice (freezes), 144 B. T. U. are given out in the process Similarly, the condensation of a pound of

steam into water at 212° liberates 966 B. T. U. This principle is applied in heating buildings by steam. The steam passes through the radiators and· condenses. The latent heat thus set free warms the building.

1987. Temperature of Mixtures.—It is often desirable to calculate the final temperature of a mixture of different substances at different temperatures. The following law is to be observed in such cases: *The quantity of heat in a mixture is the same as the quantity of heat contained in the substances before being combined.* If two substances of different temperatures are placed together, they both finally attain the same temperature; the heat lost by the one in coming from a higher to a lower temperature is gained by the other in passing from a lower to a higher temperature.

Rule.—To find the temperature of a mixture of several substances, multiply together the weight, specific heat, and temperature of each substance separately, and add the products. Next, multiply together the weight and specific heat of each of the substances separately, and add these products. Divide the former sum by the latter. The result will be the temperature of the mixture.

Let w, w_1, w_2 .... = weights of the several substances, respectively;

c, c_1, c_2 .... = specific heats of the substances, respectively;

t, t_1, t_2 .... = temperatures of the substances, respectively;

T = final temperature of mixture.

Then, $$T = \frac{w\,c\,t + w_1 c_1 t_1 + w_2 c_2 t_2 + \dots}{w\,c + w_1 c_1 + w_2 c_2 + \dots}. \qquad (137.)$$

EXAMPLE.— 15 pounds of water at 42° and 30 pounds of mercury at 70° are placed in the same vessel, and a ball of lead weighing 19 pounds and having a temperature of 110° is immersed in the mixture. What will be the final temperature of the contents?

SOLUTION.—Applying formula **137**,

$$T = \frac{15 \times 1 \times 42 + 30 \times .0333 \times 70 + 19 \times .0314 \times 110}{15 \times 1 + 30 \times .0333 \times 19 \times .0314} = 46.13°. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. A body weighing 143 pounds falls 62 feet. If the energy of the body at the end of the fall be changed into heat, how many B. T. U. will be developed ?   Ans. 11.39 B. T. U.

2. An expenditure of 210 B. T. U. per minute will develop how many horsepower ?   Ans. 4.95 H. P.

3. Supposing ¼ of the total heat of the coal to be used in doing work, how many pounds of coal must be burned per hour to run a 40 horsepower engine ? Each pound of the coal gives out 13,500 B. T. U.   Ans. 52.8 lb.

4. From what height must a block of ice fall, that the heat developed by its collision with the earth may be just enough to melt it, supposing that all of the energy gained during the fall is converted into heat ?   Ans. 112,032 feet.

5. A bar of iron weighing 20 pounds and having a temperature of 350° is plunged into a tank containing 130 pounds of water at 55°. To what temperature will the water be raised ?   Ans. 60°.

6. How many pounds of ice at 32° can be melted by 3 pounds of steam at 212° ?   Ans. 23.875 lb.

SUGGESTION.—Each pound of ice requires 144 B. T. U. to melt it; each pound of steam in changing to water at 32° gives up 1,146 B. T. U. (See Art. **1999.**)

7. How many B. T. U. are required to raise the temperature of 26 pounds of copper from 57° to 93° ?   Ans. 89.1 B. T. U.

STEAM.

PRELIMINARY IDEAS.

1988. **Steam** is *water vapor;* that is, it is *water* changed into a *gaseous state* by the application of *heat.*

The process of changing water (or other liquid) into vapor by means of heat is called **ebullition,** or **boiling.**

1989. When a vessel containing water is placed in contact with a flame of fire, the air which is generally contained in the water is first driven off and escapes from the surface without noise. The molecules of the water which are in contact with the part of the vessel nearest the fire receive heat first, and begin to move more and more rapidly until, finally, the cohesion between them is overcome, and they rise into the main body of water. At last, the whole mass of water becomes heated through, and the molecules are

then able to rise through the body of the water, overcome the pressure on the surface of the water, and escape in the form of a gas. Then the water boils.

1990. It is plain that if the pressure on the surface of the water is increased, it will take more work to force the molecules to the surface against the increased pressure. That is, more heat must be expended upon the water to make it boil, and, therefore, the boiling-point will be raised. We have seen that when water boils in open air, exposed, therefore, to the atmospheric pressure of 14.7 lb. per sq. in., the water boils when it reaches a temperature of 212°. If the pressure on the surface is increased to say 32 lb. per sq. in., the water will not boil until it reaches a temperature of 254°. On the other hand, if the pressure is lowered to 6 lb. per sq. in., the water boils when it reaches 170°. Hence, we have the following law:

An increase of pressure on the surface of a liquid raises the temperature at which it boils ; a decrease of pressure lowers the temperature at which it boils.

1991. When steam is in contact with the water from which it is generated, it is called **saturated steam.** This is the condition of steam in a boiler. According to the law just given, the temperature of saturated steam depends upon the pressure only. When the steam in a boiler shows a gauge pressure of 60 pounds, its temperature *must be* 307°. A thermometer placed in a boiler could be used to tell the pressure of the steam. It would be even more accurate (though not as convenient) than a steam-gauge.

1992. Steam, if not in contact with water, may be heated like air or any other gas until its temperature is higher than the boiling-point. For instance, let a quantity of water be placed in a cylinder as shown at *a*, Fig. 648. Suppose, for convenience, that the area of the cylinder is 100 sq. in. ; then, the pressure of the atmosphere upon the piston is 14.69 × 100 = 1,469 lb.   The number 14.69 is a little more exact than 14.7.

1993. When a part of the water is changed to steam, as shown at *b*, Fig. 648, the steam is in a saturated state, and its temperature is 212°. When, however, the water is all changed to steam, as shown at *c*, any further addition of heat will raise the temperature of the steam, while the pressure will, of course, remain at 14.69 lb. per sq. in. Steam in this condition is said to be **superheated steam.** The specific heat of superheated steam is .4805, or say .48
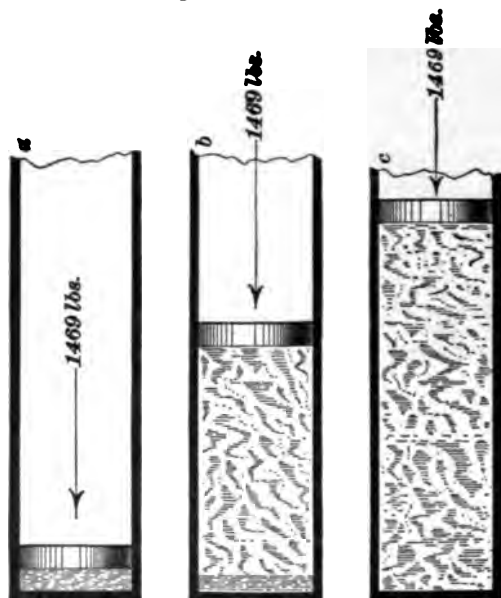
FIG. 648.

for ordinary purposes. Hence, .48 of one B. T. U. is required to raise the temperature of superheated steam one degree. The temperature of saturated steam can not be raised if the pressure remains constant. All the heat is expended in changing water to steam, and until all the water is vaporized the temperature remains constant.

1994. Prime or wet steam is steam which contains a certain percentage of water in suspension or mixed with it. If steam rises from the surface of water with a velocity greater than 2.5 feet per second, it carries water with it in

the form of spray, and when such fine spray has been once entrained or carried up with the steam, it does not readily settle against the rising current of the new steam that is constantly being formed.   Steam has been known to hold 16 times its own weight of water in suspension, or to be 1,600 per cent. moist; in the usual practice, however, the priming of steam-boilers falls within the range of from 5 to 15 per cent.

1995. Gauge and Absolute Pressures.—It has been shown that the pressure of the atmosphere is 14.7 pounds per square inch above vacuum.   Ordinary gauges register pressures above atmosphere only.   Thus, if the steam-gauge of a boiler shows 80 pounds pressure, it indicates that the pressure of the steam in the boiler is 80 pounds per square inch greater than the pressure of the atmosphere.   To find the pressure of the steam above vacuum, we must, therefore, add 14.7 .to the gauge-reading; thus, $80 + 14.7 = 94.7$.   The pressures indicated by the gauge are called **gauge pressures;** pressures above vacuum are called **absolute pressures.**   To obtain the absolute pressure, add 14.7 to the gauge pressure.

PRESSURE AND TEMPERATURE OF STEAM.

1996.   Having given the gauge pressure or the pressure above the atmosphere in a boiler, to determine the temperature of the steam and water within the boiler:

Rule.—*To 199 add 14 times the square root of the pressure.   The result will be in Fahrenheit degrees.*

Let t = temperature of steam;

p = gauge pressure of steam.

Then, $t = 199 + 14 \sqrt{p}.$ **(138.)**

EXAMPLE.—The pressure in a boiler is 81 pounds per square inch above the atmosphere, as shown by the steam-gauge; what is the temperature of the steam in the boiler ?

SOLUTION.— $t = 199 + 14 \sqrt{81} = 325°$ Fahrenheit.   Ans.

1997. Having given the temperature of the steam and water within a boiler in Fahrenheit degrees, to determine the pressure within the boiler:

Rule.—*Subtract 199 from the temperature, and divide their difference by 14. The square of this quotient will be the pressure within the boiler in pounds per square inch above the atmosphere ;*

or,
$$p = \left(\frac{t - 199}{14}\right)^2. \qquad \textbf{(139.)}$$

EXAMPLE.—The temperature of the steam within a boiler is 325° F. ; what is the pressure in the boiler ?

SOLUTION.— $p = \left(\frac{325° - 199}{14}\right)^2 = 81$ pounds per square inch above atmospheric pressure, or 81 + 14.7 = 95.7 pounds per square inch above a vacuum.   Ans.

PROPERTIES OF STEAM.

1998. The **total heat of vaporization** is the number of heat units required to change a pound of water at 32° F. to steam of the given temperature and pressure.

1999. Having given the temperature of the steam within a boiler in Fahrenheit degrees, to determine the total heat of vaporization of 1 pound of the saturated steam in the boiler from water at 32° F. :

Rule.—*Add 1,081.4 to the product of the given temperature of the steam and .305. The result will be the number of British thermal units required to convert 1 pound of water at 32° F. into 1 pound of steam at the given temperature.*

Let H = total heat of vaporization in B. T. U. ;

t = temperature of steam.

Then, $H = 1,081.4 + .305\, t.$ **(140.)**

EXAMPLE.—What is the total heat of vaporization of one pound of saturated steam at 325° F. ?

SOLUTION.— $H = 1,081.4 + .305 \times 325 = 1,180.5$ B. T. U.   Ans.

2000. The **temperature** of saturated steam does not increase by equal increments for equal advances in pressure, but rises in a decreasing ratio.   For example, at

atmospheric pressure an added pound in the steam pressure means a gain of 3.5° F. in the temperature, while at 150 pounds pressure per square inch, it means but an increase of .5° F. in temperature.

2001. The **total heat of vaporization** of steam increases but slowly with the increase in the pressure and temperature, and it takes but 1.07 times as much heat to produce a pound of steam at 485 pounds per square inch gauge pressure as it does to produce a pound under atmospheric pressure.

EXPANSION OF STEAM.

2002. Experiment has shown that when a given amount of saturated steam at a given pressure, and enclosed in a cylinder, is allowed to expand, its absolute pressure will decrease very nearly inversely as its volume increases, and that it will very closely retain its saturated state, although there will be some condensation. In other words, steam in expanding approximately follows Mariotte's law. (See Art. **852,** *Gases Met With in Mines.*)

EXAMPLE.—An engine cylinder contains 1½ cubic feet of steam at a pressure of 65.3 pounds per square inch, gauge. If the steam expands until the volume is 6 cubic feet, what will be the final gauge pressure?

SOLUTION.—Initial absolute pressure = 65.3 + 14.7 = 80 pounds per square inch.

According to Mariotte's law

$$p_1 = \frac{80 \times 1.5}{6} = 20 \text{ pounds per square inch, absolute.}$$

20 − 14.7 = 5.3 pounds per square inch, gauge. Ans.

COMBUSTION AND FUELS.

2003. **Combustion** is the *rapid* chemical combination of various *substances* with *oxygen*, as a result of which heat and light are produced.

2004. Atmospheric air is the chief source of supply from which the oxygen used in the combustion of fuels is drawn. It is composed of a mixture of oxygen and nitrogen

in the proportion of 1 pound of oxygen to 3.35 pounds of nitrogen; or, by volume, 1 cubic foot of oxygen to 3.76 cubic feet of nitrogen. Therefore, for every pound of oxygen employed in combustion, 4.35 pounds of air must be supplied, or for every cubic foot of oxygen, 4.76 cubic feet of air must be supplied. Nitrogen, however, takes no part in combustion, and, whenever present, passes off as a free gas, heated up to the temperature of the other gases with which it is mixed.

The volume of 1 pound of

Air at 62° F. is..................13.14 cubic feet.

Oxygen at 62° F. is..............11.89 cubic feet.

Nitrogen at 62° F. is.............13.50 cubic feet.

2005. **Fuels** are those forms of matter which are chiefly composed of the combustible elements, *carbon* and *hydrogen.* Coal, coke, wood, and petroleum are examples of fuels, but of these, coal is by far the most generally used in the furnaces of boilers for the production of steam.

2006. The temperature at which a combustible element or fuel takes fire, when brought into the presence of oxygen or air, differs for each substance considered, although it is a constant for any one form of matter. For example, sodium ignites and enters into chemical combination with the air at ordinary temperatures, while, in order to light an illuminating gas jet with a piece of heated iron, the iron would have to be heated to an orange color, or a temperature of about 2,000° F.

2007. Hydrogen, in whatever form it may appear, will always separate and combine with *oxygen,* when ignited, in the proportion of 1 pound of hydrogen to 8 pounds of oxygen to produce steam, in which form it will pass off and condense into 9 pounds of water; during the time it is being completely burned, 62,032 B. T. U. will be generated.

2008. The combustion of *carbon,* in like manner, is always complete at first; that is to say, 1 pound of carbon combines with 2.66 pounds of oxygen to form 3.66 pounds

TABLE 42.

One Pound of Combustible.	Theoretical Weight of Gas, in Pounds, Required to Effect the Complete Combustion of One Pound of Combustible.		Actual Weight of Air, in Pounds, Required to Effect the Complete Combustion of One Pound of Combustible.		Total Heat of Combustion of One Pound of Combustible in B. T. U.	The Equivalent of the Total Heat of Combustion, Expressed in the Number of Pounds of Water under Atmospheric Pressure it would Evaporate.	
	Oxygen.	Air.	With Chimney Draft, and Initial Temperature of Air at 62° F.	With Forced Draft at 62° F., and Waste Gases at 820° F.		From 62° F. and at 212° F.	From and at 212° F.
	1	2	3	4	5	6	7
Hydrogen	8.00	34.8	70	47	62,032	55.6	64.0
Carbon (completely burned)	2.66	11.6	22	15	14,500	13	15
Coal (of average composition)	2.46	10.7	21	14	14,133	12.67	14.63
Coke	2.50	10.9	22	15	13,550	12.14	14.02
Wood (average kiln-dried)	1.40	6.10	12	18	7,792	6.98	8.07
Petroleum	3.54	11.9	31	21	20,408	18.83	21.18

of *carbonic acid gas*; but if the supply of oxygen should be insufficient in quantity to combine with all the carbon present, and at the temperature of ignition, the carbonic acid gas will give up part of its oxygen, and reduce the final union of the two elements to the proportion of 1.33 pounds of oxygen to 1 pound of carbon to form 2.33 pounds of carbonic oxide gas.

The complete combustion of 1 pound of carbon to carbonic acid gas generates 14.500 B. T. U., but if the combustion be incomplete, that is, if the final product of the combustion is carbonic oxide gas, only 4,450 B. T. U. will be generated. If, however, this carbonic oxide gas should come in contact with more air, it will immediately ignite and combine with another 1.33 pounds of oxygen to form 3.66 pounds of carbonic acid gas again, and will regenerate the 10,050 B. T. U. which had previously been lost.

2009. In Table 42 the more important quantities that have to be considered in connection with combustibles have been tabulated; to illustrate the uses to which the table may be put, we will consider a short example.

EXAMPLE.—A furnace has a grate area of 36 square feet, upon which 453.6 pounds of coal are burned per hour, under an ordinary chimney draft. How many pounds of air must pass through the grate per minute to effect the complete combustion of the coal?

SOLUTION.—Since 453.6 pounds of coal are burned per hour, 453.6 ÷ 60 = 7.56 pounds will be consumed per minute, and from column 3, Table 42, we find that 21 pounds of air will be required per pound of coal; therefore, 7.56 × 21 = 158.76 pounds of air will have to pass through the grate per minute. Ans.

EXAMPLE.—In the last example, (a) what will be the velocity of the air through the grate, if we assume its temperature just before entering the furnace to be 62 F.; (b) what will be the total heat per hour generated by the complete combustion of the coal; (c) if no heat is lost, what amount of water will this heat evaporate from and at 212° F.?

SOLUTION.—(a) By referring to Art. **2004**, we find the volume of 1 pound of air at 62 F. to be 13.14 cubic feet; therefore, the total volume of the air that will pass through the grate per minute will be 158.76 × 13.14 = 2,086.1 cubic feet, and, dividing this by the area of the grate, we get 2,086.1 ÷ 36 = 57.95 feet per minute as the velocity of the air through the grate. Ans.

(*b*) The total heat of combustion of 1 pound of coal is 14,133 B. T. U. (see column 5, Table 42); therefore, 453.6 × 14,133=6,410,728.8 B. T. U. will be generated in the furnace per hour.   Ans.

(*c*) From column 7 of Table 42, we find the equivalent evaporation of 1 pound of coal to be 14.63 pounds of water from and at 212° F.; therefore, 453.6 pounds of coal would evaporate 453.6 × 14.63 = 6,636.168 pounds of water per hour.   Ans.

STEAM-BOILERS.

TYPES OF BOILERS.

2010. A **steam-boiler** is an apparatus for the production of steam under pressure by the expenditure of the heat energy stored in fuel.

The general principles involved in all the various boiler designs are necessarily the same, although they have assumed a variety of different forms in the effort on the part of engineers to meet the varying conditions under which boilers have to be operated.

For this reason, it has become necessary to classify them by the marked peculiarities of construction which some of the more common makes possess, and we will, therefore, take up their discussion along the natural line of their development, and under the following heads : (1) Plain Cylindrical Boilers; (2) Flue-Boilers; (3) Tubular Boilers; (4) Water-Tube Boilers.

PLAIN CYLINDRICAL BOILERS. ·

2011. A plain cylindrical boiler is simply a long hollow cylinder made of wrought-iron or steel plates riveted together, after having been bent into the required shape.   It is usually fitted with flat cast-iron heads, as shown in Figs. 649 and 650, although in some cases the heads are made hemispherical or " egg " ended, since this form offers the greatest possible resistance to bursting.

When such a boiler is in operation, the iron cylinder or shell, should be kept about two-thirds full of water, and that this may be done, a feed-water pipe *N*, leading into the boiler below the water-line *V*, Figs. 649 and 651, must be provided.
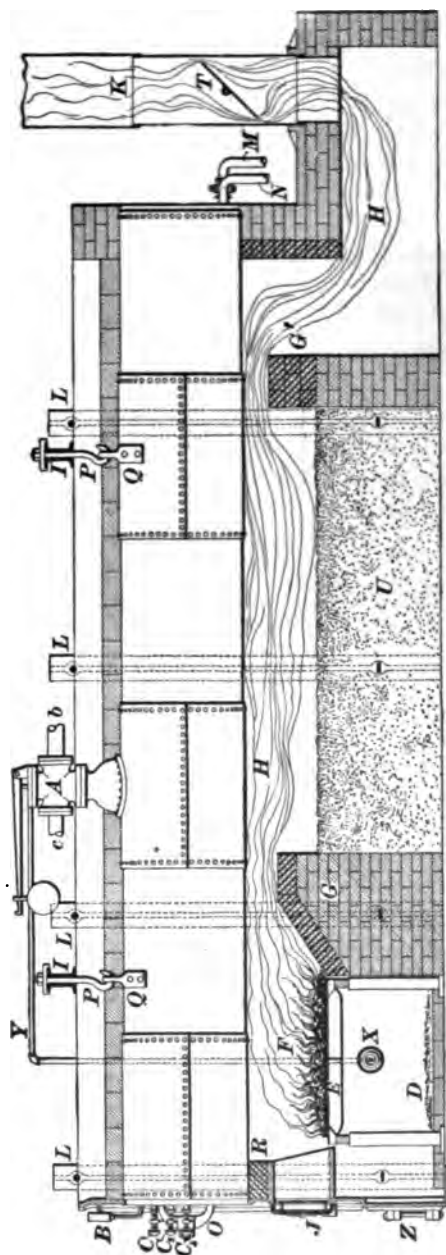
FIG. 649.

By means of this pipe, water can be forced into the boiler from time to time to supply the place of that which is evaporated into steam.

In order to be able to tell when the water in the boiler has reached its proper level, three *water-cocks* C, C_1, C_2 are placed one above the other, usually on the head of the boiler, where they will be handy to get at; they are so arranged that the middle one will come in line with the water-line V, Fig. 651, while the upper one enters the steam-space S, and the lower one the water-space W. When, therefore, the water is at its proper level, and the cocks are opened, steam should come out of the upper one, a mixture of steam and water out of the middle one, and pure water out of the lower one.

2012. The device at A is a safety-valve, a sectional view of which is given in Fig. 652. The nozzle at S communicates directly with the boiler, and the steam has a free passage through which to flow past the valve V to the steam-pipe, bolted on to the nozzle O. When, however, the steam in the boiler rises above the pressure the boiler is to carry, the valve V is lifted from its seat against the resistance offered by the lever L and weight W, through the stem P, and the steam is permitted to escape outside through the relief orifice
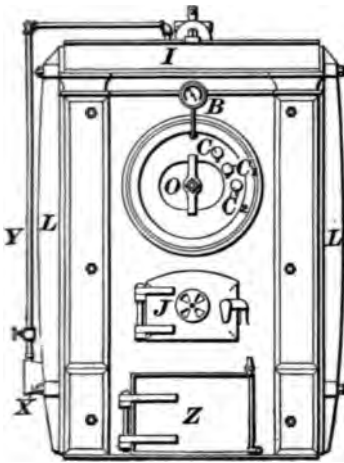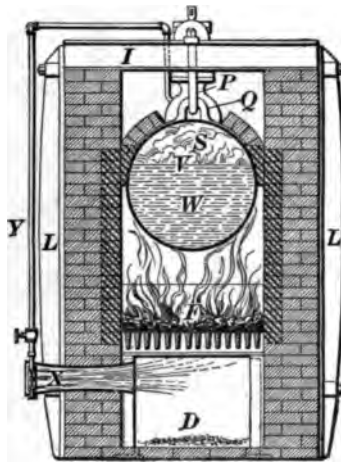
FIG. 650.	FIG. 651.

at R, until the pressure falls to its normal value. The pipe c, Fig. 649, is attached to this orifice to conduct the steam away as it escapes from the safety-valve.

The same explanation describes the second safety-valve shown in Fig. 653, which differs from the first one only in not having the additional nozzle for connecting the steam-pipe. Therefore, when this form of safety-valve is used, the steam must be led from the boiler through a pipe connected at some other point.

2013. The steam-gauge B, Fig. 650, is an instrument having a circular face and a pointer to indicate the pressure in pounds per square inch in a boiler. It should in all cases be mounted on every boiler, as it enables the engineer to see

at a glance whether the boiler is generating a greater or less
amount of steam than the circumstances require. The gauge
is connected to the boiler by a pipe.

2014. In generating the heat for the evaporation of
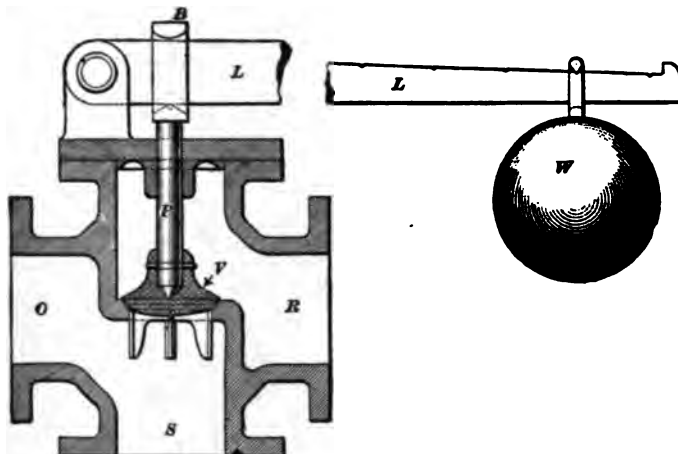
FIG. 652.
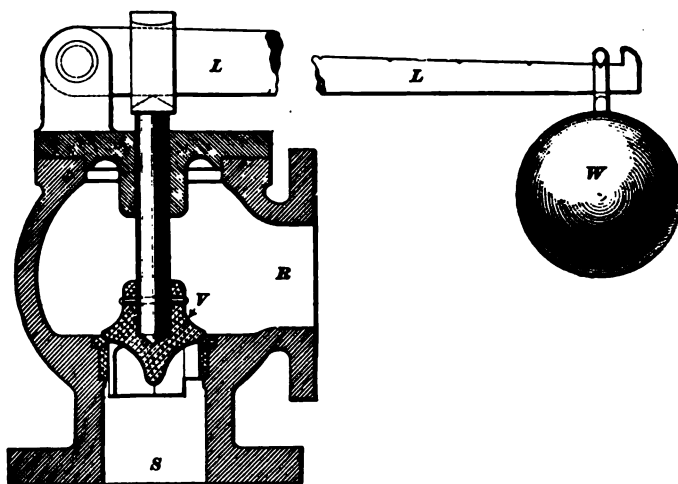
FIG. 653.

the water in these boilers, they are always **externally** fired;
that is to say, the furnace, which is made chiefly of brick·
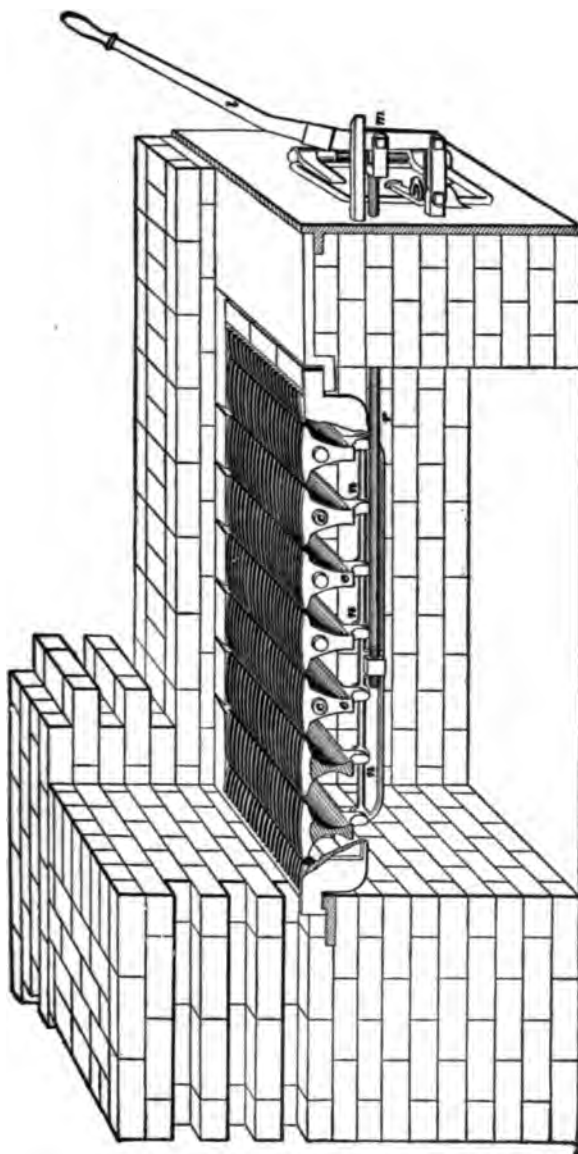
FIG. 654.

work, is built up under one end of the shell, and is also
made to form a part of the masonry enclosing the whole
boiler to prevent the heat from radiating

2015. When the boiler is in operation, the fuel which
is thrown in through the furnace-door *J* ignites and burns
on the furnace-grate *E*, Fig. 649. The furnace-grate is
frequently made up of parallel layers of cast-iron grate-bars
placed a short distance apart, which rest on iron supports
placed in the masonry. There are, however, a great many
different designs of grate-bars in use, both of the stationary
as well as of the rocking types. In Fig. 654 is shown one
of this latter kind, known as the McClave Rocking Grate.
The grate-bars *e*, *e*, *e*, made in the form of very deep **T's**,
are pivoted at both ends, and when the lever *l* is worked
backwards and forwards, the rod *r*, being connected with
l at *m*, and also with *e*, *e*, *e* at *n*, *n*, *n*, transmits the
motion of the lever to the grate-bars, and causes them
to rock backwards and forwards about their centers of
rotation *c*, *c*.

By means of these rocking grates, fires can be cleaned or
shaken down without opening the furnace-doors—a very
desirable feature, since, whenever these doors are opened,
the volume of cold air that rushes in over the grate tends
to chill the fire and lower the temperature of the furnace.

2016. Returning again to Figs. 649 and 651, the ashes
of the burning fuel fall through these grate-bars into the
ash-pit *D*, and are removed through the door *Z*, while the
hot gases generated by the burning coal pass upwards
through the combustion-chamber *F*, and are led in close
contact with the shell over the bridge-walls *G*, *G'*, and
through the flue-passages *H*, *H* to the smokestack *K*. The
ashes *U* placed beneath the boiler are for the purpose of
bringing the heated gases in contact with the bottom of the
boiler.

In order to provide for the proper cleaning of the whole
structure, a blow-off pipe *M*, through which the water may
be drained off, is led from the boiler, and a manhole *O*,

Fig. 650, closed with a manhole plate, yoke, and bolt, as shown, makes it possible for the fireman to remove the sediment and coating which from time to time are deposited in the shell by the evaporating water. Doors opening into the flues, etc., through the brickwork below the boiler also facilitate this cleaning process.

2017. Various means are employed for "setting" or supporting boilers in position. Care must be taken to so arrange the supports that the boiler-shell will be free to expand and contract with the changes of temperature.

In Figs. 649 to 651, the boiler is hung from wrought-iron channel-beams I, which rest upon the enclosing masonry work, and, whenever the plates expand or contract, the boiler swings a little on the hooks, one way or the other.

To add rigidity to the brick walls, buckstaves L, L are provided, which are bolted or keyed together above and below the boiler by long rods.

2018. In these boilers, as well as in all others, the furnace gases, when in their highly heated state, should be kept from coming in contact with those metal parts of the boiler which lie above the water-line V, since they tend, by overheating the metal, to cause a blistering of the plates and a burning off of the rivet-heads, that in time would produce serious leaks, if not an explosion. To prevent this, the masonry is made to abut against the boiler-shell just below the water-line, as seen in Fig. 651, and is frequently arched completely over the shell as well, for the purpose of diminishing the heat radiation from the metal parts of the boiler. All the masonry with which the flame does not come in contact is generally made of ordinary red brick or stone, while that with which the flame does come in contact is made of firebrick.

2019. The draft or rapidity with which the air flows through the grate of a boiler, for the purpose of supplying the fuel with a sufficient quantity of oxygen to insure its complete combustion, is usually produced by the chimney

or smokestack, although it is frequently increased and made more efficient by connecting a blower with the ash-pit *D*.

There are a great many different kinds of these blowers, but the simplest and the one best adapted for boiler work is that represented in Figs. 649 to 651, at *X*. It consists simply of a long metal cylinder into which a jet of steam is led from the boiler by a ¾-inch pipe.

The steam, as it rushes through the pipe *Y* into the blower with great velocity, draws the air along with it, and the cylinder, by giving the blast the proper direction, causes it to impinge on the grate-bars *E* ; thus a rapid and complete combustion of the coal is produced.

2020. Plain cylindrical boilers are little used at the present day, except in mining districts and other localities where fuel is very cheap, for they have so small a water-heating area, in proportion to the amount of water they contain, and the volume of gas given off from their furnaces, that they are very wasteful of heat energy. They are made from 28 to 50 inches in diameter, and from 20 to 60, and even 100, feet in length. This great length is given to increase the water-heating area.

FLUE-BOILERS.

2021. The flue-boiler represents a type in which an increased water-heating area is obtained by the introduction of one or two large flue-pipes within the boiler-shell, below the water-line. In Figs. 655 to 657 is shown an *externally fired flue-boiler*, or one in which the heated gases, after passing from the furnace, over the bridge-walls, and along in contact with the lower surface of the boiler till the space *H* is reached, are made to return through one or two large flues *A, A*, Fig. 657, fitted within the cylinder below the water-line.

From these flues the gases enter the smokebox *B*, and flow from there directly into the smokestack *C*. The arrangement of the masonry and the "setting" of the boiler-shell, in this instance, follows the construction of Figs. 649 to 651 so closely that no further explanation need be
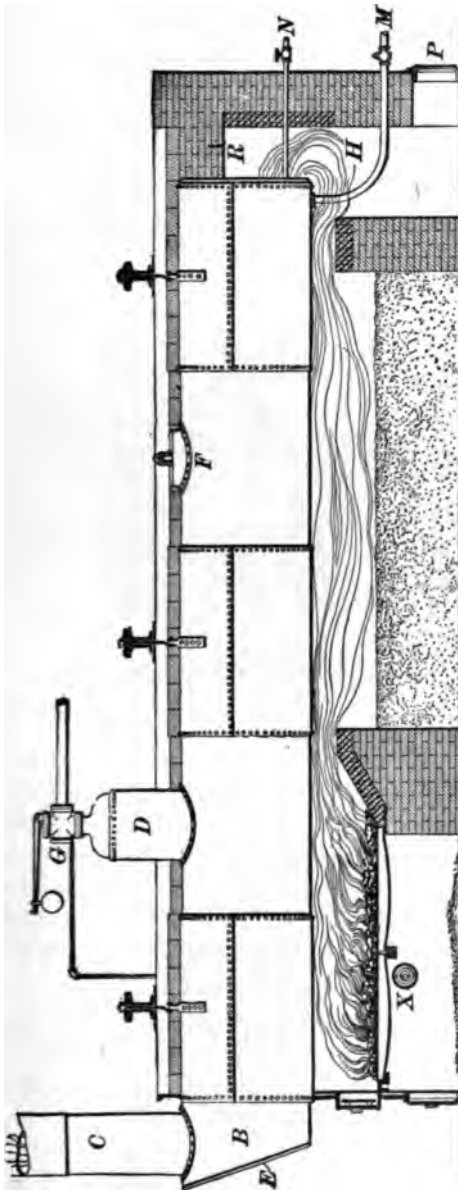
FIG. 685.

given, other than to call attention to the feed-pipe at *N*, the blow-off pipe at *M*, the steam-gauge at *K*, the gauge-cocks on the column *L*, and the steam-dome placed above the boiler at *D*. The steam-gauge and the gauge-cocks communicate with the boiler through the pipes *s* and *t*, the former passing into the steam-space and the latter into the water.

2022. There is generally a steam-dome on every boiler, which serves as a chamber in which the steam collects and is dried or relieved of a portion of its entrained water before passing to the engine. The hole in the shell of the boiler, over which the steam-dome is riveted,

should not be made larger than is sufficient to permit the free passage of the steam. Anything greater than this only weakens the shell, without adding to the utility of the dome.

To facilitate the cleaning of the flues and boiler, a door is provided in the smokebox at *E*, and a manhole in the shell at *F*. A door should also be made in the masonry work to enable the fireman to get at the external flues of
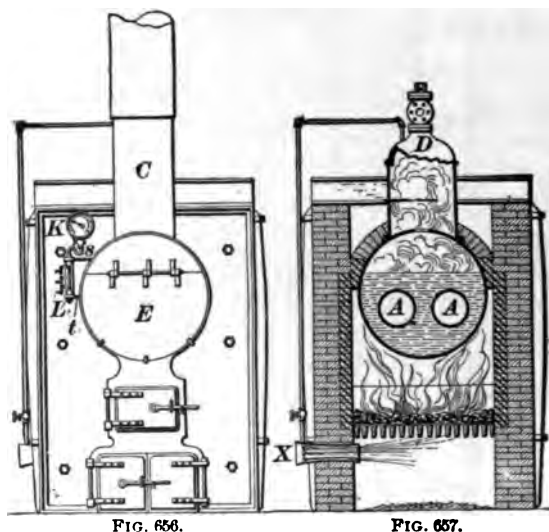
FIG. 656.   FIG. 657.

the structure. Access is given to the rear end of the shell and to the pipes *M* and *N* through the door *P*. The cast-iron plate *R* supports the brickwork over the space *H*. The furnace of this boiler is supplied with the same form of steam-jet blower *X* as that described in connection with Fig. 649.

2023. In Fig. 658 a design of an *internally fired flue-boiler* is shown.

In this type, the furnace is entirely surrounded by water, and the bridge-wall *D*, the grate *C*, and the ash-pit *P* are placed within the flue. If these flues (only one of which is shown in the figure) are corrugated, their capacity for resist-

ing external pressure is greatly increased over that of the plain flue; they are, consequently, much less liable to collapse or to be pushed in. Above the flue, and surrounded by water, are a large number of tubes leading from the chamber E through the shell to the smokebox F, which are provided to convey the heated furnace gases a second time through the water after they have traversed the corrugated flue.

The tubes B, besides greatly increasing the heating surface of the boiler, combine with the flue and stayrods G to
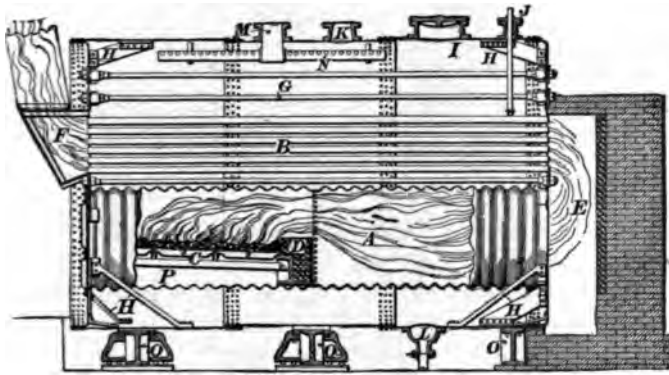
FIG. 658.

increase the strength of the boiler-heads, since when the boiler is in operation the internal pressure puts them under a tensional strain. Further stiffness is given to the boiler by riveting a number of plates H, H (called *gusset-stays*) around the head and inner surface of the shell.

To facilitate the inspection and cleaning of the interior of the shell, a cast-iron manhole I, closed by a bolted cover, is riveted on the upper surface of the boiler, and for cleaning the tubes B a door is provided to the smokebox F. The smokestack is led from the opening in the top of the smokebox F.

The setting of this boiler is somewhat different from those previously considered. The masonry is simply built up to prevent heat radiation, and to insure the flow of the furnace

gases through the return tubes *B* of the boiler.   The curved beams *O*, *O*, *O* support the boiler.

The steam-gauge and water-cocks are not shown in this figure.   The safety-valve should be bolted on at *K*, and the blow-off pipe at *L*.   The feed-pipe *J* leads into the steam-space, and discharges below the tubes *B*.   The steam supply is drawn from the nozzle *M*, through the pipe *N*.   The pipe *N* is made to take the place of a steam-dome, since the steam, in passing through the small holes of the pipe, is freed from the greater part of its entrained water.

Flue-boilers, when properly designed and constructed, give very good working results, and have given good satisfaction among English engineers.   In America, they are not so popular as some of the other makes, although they are found in operation at a number of plants.

TUBULAR BOILERS.

2024.   Tubular boilers, as in the case of flue-boilers, may be divided into two classes, the *internally fired* and *externally fired*.   The internally fired boiler shown in Fig. 659
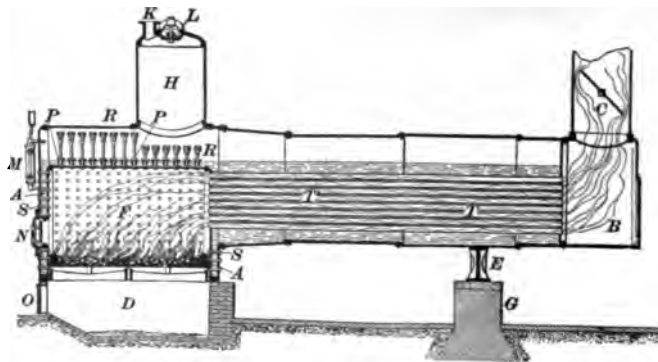
FIG. 659.

is called the **fire-box or locomotive boiler.**   This type of boiler, next to the multitubular boiler, is probably more used than any other type.   It is used exclusively in railway service, and also largely as a stationary boiler.   A large pro-

portion of the small portable combined engines and boilers used for agricultural purposes have the fire-box type of boiler. The general construction of this type of boiler is shown in Fig. 659. The shell is composed of two differently shaped parts riveted together. The front part of the shell is cylindrical; the rear part is usually of a rectangular cross-section with vertical sides, or of a trapezoidal section with inclined sides; in either case, the top is semicylindrical. The furnace F is a box of the same shape as the rear end of the shell in which it is placed. There is a space left between the sides and end of the furnace and the shell; this space is filled with water, as shown at A, A. A series of tubes extend from the front sheet of the furnace or fire-box to the front head of the shell. The shell is prolonged beyond the front head, forming a smokebox B, into which opens the stack C.

As shown in this figure, the *water-legs* (as the spaces A, A are called) only extend down as far as the grate, the ash-pit D being formed in the brick-setting. In many boilers of this type, the water-legs extend down to the bottom of the ash-pit, and sometimes there is a water-space below the ash-pit; that is, the furnace and ash-pit are entirely surrounded by water.

The boiler is supported at the front end by the cast-iron cradle E resting upon the masonry foundation G. The rear end is supported upon a brick wall, which also forms the ash-pit. The boiler is usually provided with a dome H, from which is led the main steam-pipe, which is bolted on at K. In the figure, the dome is provided with a manhole L. The feed-water may be introduced at any convenient point in the shell. The pressure-gauge, water-glass, and gauge-cocks are attached to the column M, which is placed in communication with the interior of the shell. The furnace and ash-pit doors are shown at N and O, respectively. The safety-valve is usually attached to the dome.

Since the flat sides of the furnace and shell are liable to bulge on account of the pressure, they must be braced or stayed. This is accomplished by the staybolts S, S. The

24—10

flat top of the fire-box is strengthened by a series of parallel
girders *P*, *P*. As an additional security, the girders are
sometimes attached to the shell by the "sling-stays" *R*, *R*.

The gases of combustion pass directly from the furnace
through the tubes *T*, *T*, to the smokebox *B*, and out of the
stack *C*. In railway locomotives, a strong draft is obtained
by allowing the exhaust steam to discharge through the
smokestack. The escaping steam carries along the air and
the escaping gases in the smokebox *B*, thereby drawing a
new supply of gases through the tubes *T*, *T*, and a supply
of air through the grate.

The tubes of the locomotive boiler are about 12 feet long,
two inches in diameter, and made of iron or steel. The tubes
of stationary and portable boilers of this type are generally
of larger diameter, as there is less demand for great quanti-
ties of steam. The locomotive type of boiler is **self-con-
tained ;** that is, it requires no brickwork for flues or for
setting.

2025. The Return-Tubular Boiler.—This type of
boiler is a development of the flue-boiler, the two large flues
of the latter being replaced by a large number of small
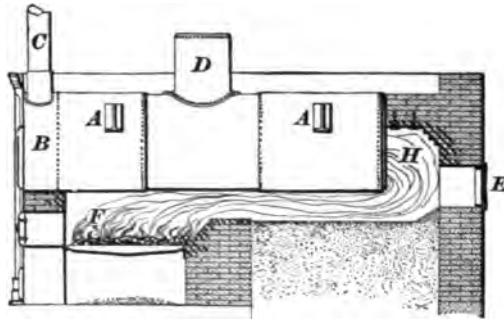
FIG. 660.

tubes. The object of introducing the numerous tubes is to
increase the heating surface of the boiler.

A side view of a tubular boiler is shown in Fig. 660; a
cross-section through the tubes is shown in Fig. 661. The

tubes extend the whole length of the shell; the ends are
beaded into holes in the heads of the boiler.   The front end
of the shell projects beyond the head,
forming the smokebox *B*, into which
opens the stack *C*.

The shell is suspended on the side
walls by the brackets *A*, *A*, which are
riveted to the shell.   The boiler is usual-
ly provided with a dome *D*, though this
is sometimes left off.   The walls are
built and supported by buckstaves in
practically the same manner as those
previously described.   Since this type of   Fig. 661.
boiler is generally short, one bridge only is used.   Firebrick
is used for all parts of the wall exposed to the fire or heated
gases.   The fittings are not shown in the figure.   The safety-
valve would be placed on top of the dome, and the pressure-
gauge and gauge-cocks would be placed on the front.
The manhole is either in one of the heads or on top of the
shell.   The feed-pipe may enter the front head or the top,
while the blow-off pipe is placed at the bottom of the shell,
at the rear end.   Access is given to the rear end of the boil-
er through the door *E*.

As usual, the furnace *F* is placed under the front end of
the boiler.   The gases pass over the bridge, along under the
boiler into the chamber *H*, then back through the tubes to
the smokebox *B*, and out of the stack *C*.

The return-tubular boiler is probably more used in the
United States than any other.   The details of its construc-
tion and setting will be shown later.

2026.   The Vertical Boiler.—This type is essentially
a modification of the locomotive type placed on end.   A com-
mon form of vertical boiler is shown in Fig. 662.   It con-
sists of a vertical cylindrical shell, in the lower end of which
is placed a fire-box *F*.   The lower rim of the fire-box and the
lower end of the shell are separated by a wrought-iron
ring *k*, to which both are riveted, the rivets going through

both plates and ring.   The shell and fire-box are also stayed together by the staybolts *a, a.*   The space between the two is filled with water, so that the fire-box is nearly sur-rounded by it.   The boiler-shell, and likewise the grate *E,* rest upon a cast-iron base *D* which forms the ash-pit.   A series of vertical tubes *t, t* ex-tend from the top sheet of the fire-box to the upper head of the shell.   The tubes serve as stay-rods and strengthen the flat surfaces which they connect. The upper ends of the tubes open directly into the chim-ney or smokestack *K.*   The gases from the furnace thus pass directly through the tubes and out of the stack.
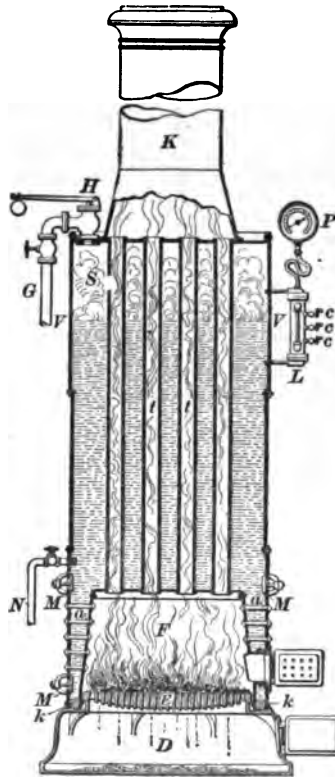
The safety-valve is shown at *H,* with the main steam-pipe *G* leading from it.   The pres-sure-gauge *P* and gauge-cocks *c, c, c* are attached to a column *L,* which communicates in the usual manner with the interior of the shell.   The construction of this type of boiler does not generally permit the use of

FIG. 662.

manholes, but handholes *M, M* are placed in convenient positions for cleaning out mud and sediment.

The boiler is fed through the feed-pipe *N,* which is con-nected to a pump or injector.

When the tubes extend through to the upper head of the boiler, as shown in Fig. 662, their upper ends pass through the steam-space *S* above the water-line *V V.*   This is con-sidered to be a bad feature, since the tubes are liable to

become overheated and to collapse. when the boiler is subject to rapid firing.

In the form of vertical boiler shown in Fig. 663, this danger is avoided.   A chamber or smokebox *I* extends down from the upper head of the shell so that its bottom plate is always below the water-line.   The upper ends of the tubes *t, t* are expanded into the lower plate of this chamber, and, therefore, the tubes are always surrounded by water from end to end.   A vertical boiler constructed in this manner is said to have a *submerged head.*   Aside from the submerged head, the construction of the boiler of Fig. 663 is similar to that of Fig. 662.

Vertical boilers are generally wasteful of fuel; they are, however, self-contained, require but little floor space, and are easy to construct and repair.   For these reasons, the vertical type of boiler is very popular with a large class of steam-users.
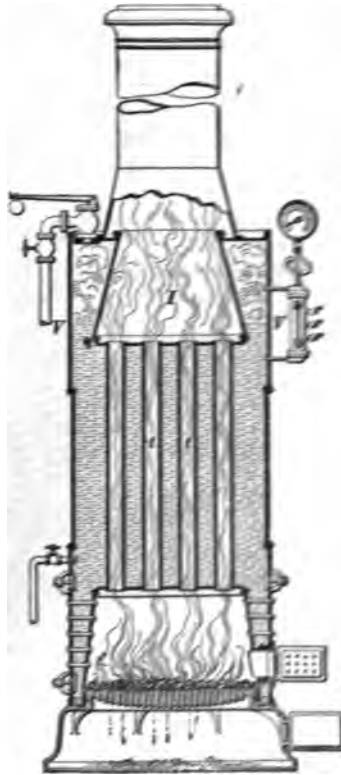
FIG. 663.

WATER-TUBE BOILERS.

2027.   The **Babcock and Wilcox water-tube boiler** is shown in Fig. 664.   It consists essentially of a main horizontal drum *B* and of a series of inclined tubes *T, T*.   (Only a single vertical row of tubes is shown by the figure, but it will be understood that each nest of tubes is composed of several vertical rows.)   There are usually 7 or 8 of these vertical rows to each horizontal drum.   The front ends of the tubes of a vertical row are all expanded into a hollow iron

casting H called a **header.** The rear ends are expanded into a similar header, and the front and rear headers are placed in communication with the drum by tubes, or *risers*, C and C', respectively.   In front of each tube, a handhole is placed in the header for the purpose of cleaning, inspecting, or removing the tubes.

The method of supporting the boiler is not shown in the figure.   The usual method is to hang the boiler from wrought-iron girders resting on vertical iron columns. The
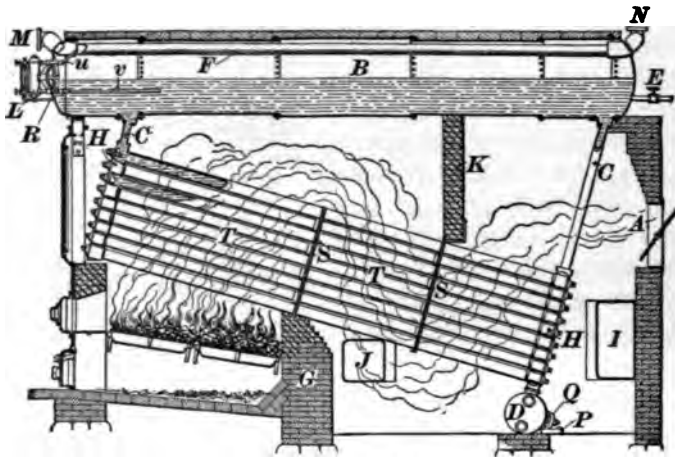
FIG. 664.

brickwork setting is not depended upon as a means of support.   This make of boiler, in common with all others of the water-tube type, requires a brickwork setting to confine the furnace gases to their proper field.

The furnace is of the usual form, and is placed under the front end of the nest of tubes.   The bridge-wall G is built in contact with the tubes; another firebrick wall K is built between the tubes and drum.   These walls and the baffle-plates S, S force the hot furnace gases to follow a zigzag path back and forth through the tubes.   The gases finally pass through the opening A in the rear of the wall, into the chimney-flue.

The feed-water is introduced through the feed-pipe E. The steam is collected in the dry-pipe F, which terminates

in the nozzles M and N, to one of which is attached the main steam-pipe, and to the other the safety-valve.

The pressure-gauge, cocks, etc., are attached to the column, which communicates with the interior of the shell by the small pipes u and v, the former of which extends into the dry-pipe, the latter into the water.

At the bottom of the rear row of headers is placed the mud-drum D.   Since this drum is the lowest point of the water-space, most of the sediment naturally collects there. This sediment may be blown out from time to time through the blow-out pipe P.   The drum D is provided with a hand-hole Q, and a manhole R is placed in the front head of the drum B.   The heads of the drums are of hemispherical form, and, therefore, do not require bracing.   Access may be had to the space within the walls through the doors I and J.

The circulation of water takes place as follows: The cold water is introduced into the rear of the boiler; the furnace being under the higher end of the tubes, the water in that end expands upon being heated, and is also partially changed to steam; hence, a column of mingled water and steam rises through the front headers to the front end of the drum B, where the steam escapes from the surface of the water.   In the meantime, the cold water fed into the rear of the drum descends to the rear headers through the tubes C, to take the place of the water which has risen in front. Thus, there is a continuous circulation in one direction, sweeping the steam to the surface as fast as it is formed, and supplying its place with cold water.   Most of the sediment sinks to the mud-drum D, from which it is blown out from time to time.

2028.   The **Heine water-tube boiler** is shown in Fig. 665.   It differs in many respects from those already described.   It consists of a large main drum A, which is above and parallel with the nest of tubes T, T.   Both drum and tubes are inclined at a small angle with the horizontal, so that the water-level is about $\frac{1}{3}$ the height of the drum in front and about $\frac{2}{3}$ the height in the rear.   The ends of the

tubes are expanded into the large wrought-iron **water-legs**
B, B.   These legs are flanged and riveted to the shell.   The
shell is cut out for about ⅛ the circumference to receive the
water-legs, the opening being from 60 to 90 per cent. of the
cross-sectional area of the tubes.   The drum-heads are of a
hemispherical form, and, therefore, do not need bracing.

The water-legs form the natural support of the boiler.
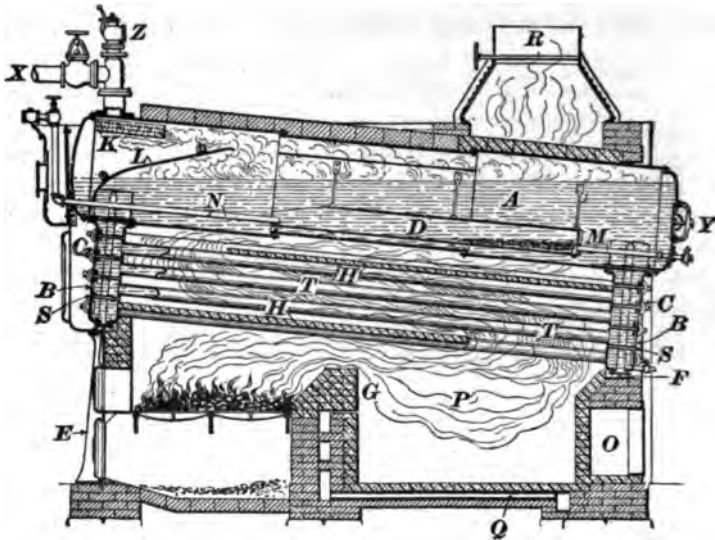
FIG. 665.

The front water-leg is placed on a pair of cast-iron columns
E which form part of the front of the boiler.   The rear water-
leg rests on rollers (shown at F) which may move freely on
a cast-iron plate bedded in the rear wall.   The rollers allow
the boiler to expand when heated.

The boiler is enclosed by a brickwork setting in the usual
manner.   The bridge G is made largely of firebrick.   It is
made hollow, and has openings in the rear to allow air to pass
into the chamber P and mix with the furnace gases.   The air
is drawn from the outside through the channel Q in the side
wall.   The air is, of course, heated in passing through the
bridge.   In the rear wall is the arched opening O, which is

closed by a door, and further protected by a thin wall of firebrick.   When it is necessary to enter the chamber P, the wall may be removed and afterwards replaced.

The feed-water is brought in through the feed-pipe N, which passes through the front head.   As the water enters, it flows into the mud-drum D, which is suspended in the main drum below the water-line, and is thus completely submerged in the hottest water in the boiler.   This high temperature is useful in precipitating the impurities contained in the feed-water.   These impurities settle in the mud-drum D, and may then be blown out through the blow-out pipe M.

Layers of firebrick H, H are laid at intervals along the rows of tubes, which act as baffle-plates, and force the furnace gases to pass back and forth through the tubes.   The gases finally escape through the chimney R placed above the rear end of the boiler.   To protect the steam-spaces of the drum from the action of the hot gases, the drum in the vicinity of the chimney is protected by firebrick, as shown in the figure.

The steam is collected and freed from water by the perforated dry-pipe K.   The main steam-pipe with its stop-valve is shown at X, the safety-valve at Z.   In order to prevent a combined spray of mixed water and steam from spurting up from the front header and entering the dry-pipe, a deflecting plate L is placed in the front end of the drum.

A manhole Y is placed in the rear head of the drum. The flat sides of the water-legs, which are made hollow to give access to the outside of the tubes, are stayed together by the staybolts S, S.   In front of each tube, a handhole C is placed to give access to the interior of the tubes.

Where a battery of several of these boilers is used, an additional steam-drum is placed above and at right angles to the drums A, A.

2029.   The **Stirling boiler,** shown in Fig. 666, is a departure from the regular type of water-tube boilers.   It consists of a lower drum A, connected with three upper

drums B, B, B by three sets of nearly vertical tubes. These upper drums are in communication through the curved tubes C, C, C. The curved forms of the different sets of tubes allow the different parts of the boiler to expand and contract freely without strain.

The boiler is enclosed in a brickwork setting, as shown.
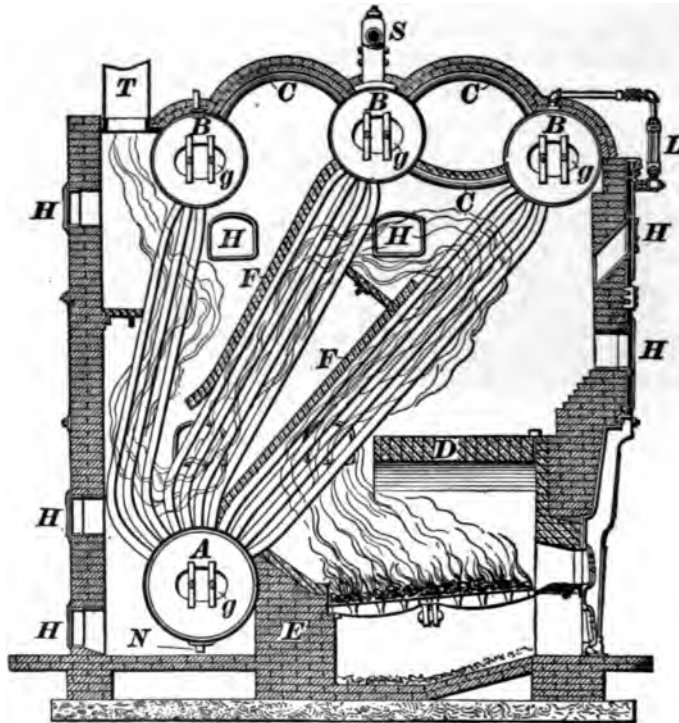
FIG. 666.

The setting is built with various holes H, H, so that the interior may be inspected or repaired.

The boiler is suspended from a framework of wrought-iron girders, not shown in the figure.

The bridge E is lined with firebrick, and is built in contact with the lower drum A and the front nest of vertical tubes. An arch D is built above the furnace, and this, in

connection with the bafflers F, F, directs the course of the heated gases, causing them to pass up and down through the tubes. The arch and bafflers are made of firebrick.

The cold feed-water enters the rear upper drum and descends through the rear nest of tubes to the drum A, which acts as a mud-drum, and collects the sediment brought in by the water. A blow-off pipe N permits the removal of the sediment. The steam collects in the upper drums B, B. To the middle drum is attached the steam-pipe and safety-valve S.

The chimney T is located behind the rear upper drum. Therefore, the cold feed-water enters the coolest part of the boiler, and the circulation of the water is directly opposite to that of the escaping hot gases.

The water-column L with its fittings is placed in communication with the front upper drum. All the drums are provided with large manholes g.

The boiler is made with a cast-iron front.

2030. The following advantages are claimed for the Stirling boiler:

(1) The vertical position of the tubes prevents the collection of sediment, and at the same time encourages the rapid rise and separation of the steam as soon as it is formed. (2) The boiler is very simple and easy to construct; there are no flat surfaces to be stayed, and there is little or no machine work required in its manufacture. (3) It is very accessible for cleaning or repairs; any part of the boiler may be inspected by removing the four manhole plates g.

The various water-tube boilers just described are coming into extensive use. The most important points in their favor are their safety from disastrous explosion and their economy in the use of fuel. An objection sometimes urged against water-tube boilers is that they require more attention; since they usually have much less cubic capacity than cylindrical boilers of the same power, the water-level must be closely watched.

STRENGTH OF BOILERS.

2031. Steam-boilers can be designed and constructed to safely generate and operate under almost any desired steam pressure, however great it may be. The common practice among engineers, however, is rarely, if ever, to go above 250 pounds per square inch, and in the majority of plants throughout the country the steam pressure does not exceed 60 pounds per square inch.

2032. In approximately determining the safe working pressure under which any well-designed boiler may be operated, it is only necessary to find the diameter of the largest cylindrical shell used in its construction, and the thickness of the plate of which the shell is made. Then the safe working pressure may be found by the following rule:

Rule.—*Multiply the thickness of the plate in inches by the constant given below, and divide the product by the diameter of the shell in inches; the quotient will be the allowable gauge pressure.*

Let p = safe working pressure;
  t = thickness of plate in inches;
  d = diameter of shell in inches;
  c = constant.

Then, $p = \dfrac{c\,t}{d}.$ **(141.)**

The following constants are to be used in formula **141 :**

Wrought-iron plate, single-riveted joint............ 10,224
Wrought-iron plate, double-riveted joint........... 13,152
Steel plate, single-riveted joint.................... 16,608
Steel plate, double-riveted joint.................. 20,688

EXAMPLE.— If a return-tubular boiler is made of $\frac{5}{16}$ of an inch thick wrought-iron boiler plate, double riveted, and is 5 feet in diameter, what is the greatest steam pressure under which such a boiler can be safely operated?

SOLUTION.—Applying formula **141,**

  $p = \dfrac{13,152 \times \frac{5}{16}}{60} = 68.5$ pounds per square inch, **gauge. Ans.**

68.5 + 14.7 = 83.2 pounds per square inch above a **vacuum.**

HORSEPOWER OF BOILERS.

2033. The **horsepower** of a boiler is a measure of its capacity for generating steam. Boiler-makers usually rate the horsepower of their boilers as a certain fraction of the heating surface; but this is a very indefinite method, for with the same heating surface, different boilers of the same type may, under different circumstances, generate different quantities of steam.

In order to have an accurate standard of boiler-power, the American Society of Mechanical Engineers has adopted as a standard horsepower *an evaporation of 30 pounds of water per hour from a feed-water temperature of 100° F. into steam at 70 pounds gauge pressure*, which is considered equivalent to 34.5 units of evaporation; that is, to 34.5 pounds of water evaporated from a feed-water temperature of 212° F. into steam at the same temperature.

EXAMPLE.—A boiler evaporates per hour 1,980 pounds of water from a feed temperature of 100° into steam at 70 pounds gauge pressure. What is the horsepower of the boiler?

SOLUTION.—Since, under the given conditions, an evaporation of 30 pounds is equivalent to one horsepower, the number of horsepower is $1,980 \div 30 = 66$. Ans.

2034. In the various types of boilers there is a nearly constant ratio between the water-heating surface and the horsepower, and also between the heating surface and the grate area. These ratios are given in the following table:

TABLE 43.

RATIO OF HEATING SURFACE TO HORSEPOWER AND OF HEATING SURFACE TO GRATE AREA.

Type of Boiler.	Ratio = $\dfrac{\text{Heating Surface}}{\text{Horsepower}}$	Ratio = $\dfrac{\text{Heating Surface}}{\text{Grate Area}}$
Plain Cylindrical	6 to 10	12 to 15
Flue	8 to 12	20 to 25
Return-Tubular.	14 to 18	25 to 35
Vertical	15 to 20	25 to 30
Water-Tube	10 to 12	35 to 40
Locomotive	1 to 2	50 to 100

If the heating surface of a boiler is known, the horse-power can be found roughly; thus, if a return-tubular boiler has a heating surface of 900 square feet, its horsepower lies between $\frac{900}{18} = 50$ H. P. and $\frac{900}{14} = 64.3$ H. P., say about 57 H. P.

2035. The *heating surface* of a boiler is the portion of the surface exposed to the action of flames and hot gases. This includes, in the case of the multitubular boiler, the portions of the shell below the line of brickwork, the exposed heads of the shell, and the interior surface of the tubes. In the case of a water-tube boiler, the heating surface comprises the portion of the shell below the brickwork, the outer surface of the headers, and outer surface of tubes. In any given case, the heating surface may be calculated by the rules of mensuration. The following example will show the method of calculating the heating surface of a return-tubular boiler :

EXAMPLE.—A horizontal return-tubular boiler has the following dimensions: Diameter, 60 inches; length of tubes, 12 feet; internal diameter of tubes, 3 inches; number of tubes, 82. Assume that $\frac{3}{4}$ of the shell is in contact with hot gases or flame and $\frac{2}{3}$ of the two heads are heating surface.

SOLUTION.—

Circumference of shell $= 60 \times 3.1416 = 188.496 = 188.5$ in., say.

Length of shell $= 12 \times 12 = 144$ in.

Heating surface of shell $= 188.5 \times 144 \times \frac{3}{4} = 18,096$ sq. in.

Circumference of tube $= 3 \times 3.1416 = 9.425$ in., nearly.

Heating surface of tubes $= 82 \times 144 \times 9.425 = 111,290.4$ sq. in.

Area of one head $= 60^2 \times .7854 = 2,827.44$ sq. in.

Two-thirds area of both
  heads $= \frac{2}{3} \times 2 \times 2,827.44 = 3,769.92$ sq. in.

From the heads must be subtracted twice the area cut out by the tubes; this is $82 \times 3^2 \times .7854 \times 2 = 1,159.26.$

Total heating surface in square feet =

$$\frac{18,096 + 111,290.4 + 3,769.92 - 1,159.26}{144} = 916.64 \text{ sq. ft.} \quad \textbf{Ans.}$$

CHIMNEYS.

2036. Chimneys have two important duties to perform, the first being to carry off the waste furnace gases, which requires size, and the second, to produce a draft sufficient to insure the complete combustion of the fuel, which requires

height. The area of a chimney is usually made from one-seventh to one-tenth as large as the area of the furnace-grates, or of about the same cross-section as the cross-sectional area of the flues or tubes; we have, therefore, a comparatively simple method of determining one of the required dimensions of a chimney, and, when this is known, it becomes an easy matter to determine the height of the chimney when the horsepower of the boiler has been ascertained.

The horsepower of a boiler being given and the necessary chimney area having been determined, the following rule gives the required height that the chimney must be to produce the necessary draft :

Rule.—*From 3.33 times the area of the chimney in square feet, subtract twice the square root of the area of the chimney in square feet, and divide the given horsepower by the remainder. The square of the quotient will be the height of the chimney in feet.*

Let A = area of chimney;
  H = horsepower of boiler;
  h = height of chimney.

Then, $h = \left(\dfrac{H}{3.33\,A - 2\sqrt{A}} \right)^{2}.$ **(142.)**

EXAMPLE.—What must be the height of a chimney which is to have a cross-sectional area of 7 square feet, and to supply the draft for a 141 horsepower boiler ?

SOLUTION.—Using formula **142,**

$h = \left(\dfrac{141}{3.33 \times 7 - 2\sqrt{7}} \right)^{2} = \left(\dfrac{141}{3.33 \times 7 - (2 \times 2.65)} \right)^{2} = 61.3$ feet.   Ans.

STEAM-ENGINES.

THE PLAIN SLIDE-VALVE ENGINE.

GENERAL DESCRIPTION.

2037. The **plain slide-valve engine** is the most simple of all of the many forms of steam-engines now in use. In its construction and operation, however, all of the fundamental principles of this class of machinery are involved.

2038. In Fig. 667 such an engine is shown, and in Fig. 668 is shown an enlarged section of a steam-cylinder. Referring to these figures, H is the head end and C the crank end of the steam-cylinder; B and B' are the steam-ports; D is the steam-chest; E is the exhaust-port; N and N' are the cylinder-heads; S is the steam supply-pipe; O is the exhaust-pipe, and connects with the exhaust-port E; G is one of the two guide-bars (the other, which is not designated, is on the opposite side of the cross-head *2*); R and R' are the shaft-bearings, and T is the bed or frame of the engine. The above are all stationary parts of the engine, or parts which do not change their relative positions when the engine is in motion. P is the piston; *1* is the piston-rod; *2* is the cross-head; *3* is the cross-head pin; *4* is the connecting-rod; *5* is the crank; *6* is the crank-pin; *7* is the crank-shaft; *8* is the fly-wheel; *9* is the eccentric; *10* is the eccentric-strap; *11* is the eccentric-rod; *12* is the rocker; *13* is the valve-rod, or stem, and V is the slide-valve. These are all movable parts of the engine, or parts which change their relative positions when the engine is in motion.

2039. The stroke of the engine is equal to the throw of the crank, or to the diameter of the circle described by the

§ 19
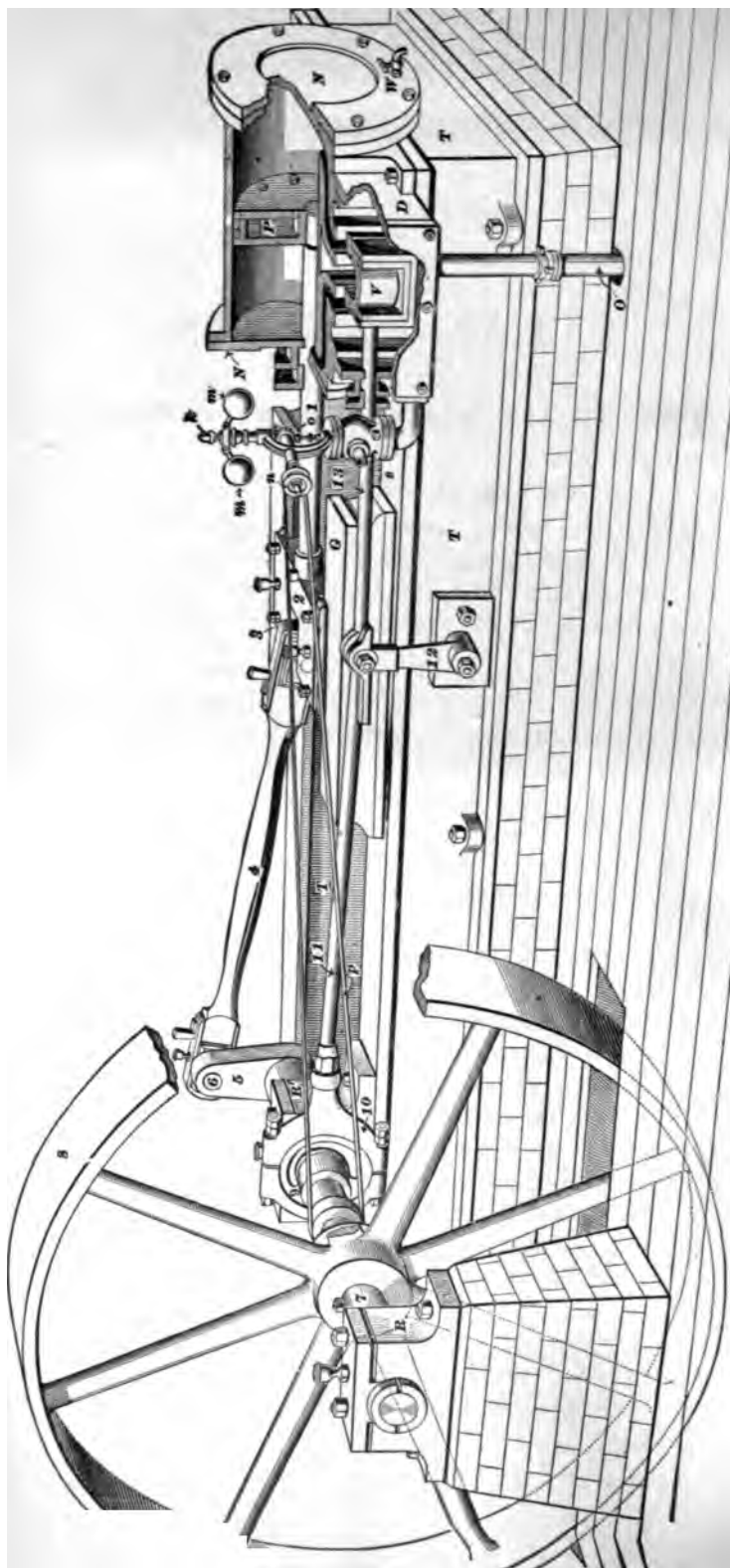
FIG. 687.

center of the crank-pin 6.   It is also equal to the cross-head
or piston travel, or the distance through which the cross-
head or piston moves, and it determines the working length
W of the cylinder, as shown in Fig. 668.   The bore of the
cylinder is M, Fig. 668.   The counter-bores F and F' are
enlargements, into which the piston projects at the end of
each stroke.   They prevent the formation of shoulders at
the ends of the cylinder by insuring an equal wear of the
cylinder over its entire working length.   Such shoulders
would cause a pounding of the piston when the length of the
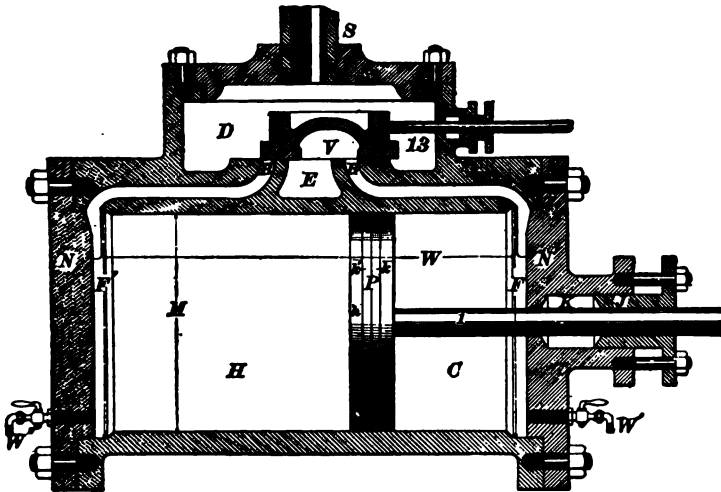
FIG. 668.

connecting-rod is increased by the taking up of the wear of
its joints.   The clearance at one end of the cylinder is the
volume that remains when the piston has completed its
stroke—it includes the steam-port.   It is diminished at the
crank end by the volume of that portion of the piston-rod
remaining within the cylinder.   The volume of the cylinder
is equal to the volume of the clearance at one end plus the
volume swept through during one stroke of the piston.   It
is less at the crank end by an amount equal to the volume
of that portion of the piston-rod remaining in the cylinder.

Drain-valves *W'* and *W''* are fitted in each end of the cylinder through which any condensed steam may be discharged.

The piston is given a loose fit in the cylinder, and has split rings *k* and *k'* inserted, which spring out so as to press against the wall of the cylinder, and prevent leakage of steam between the wall of the cylinder and piston. Pistons are usually supplied with a follower-plate *h*, which is bolted to the head end of the piston *P*, in order to hold these split rings *k* and *k'* in place. The piston-rod *1* is a perfectly round, smooth bar, rigidly connected to both the piston *P* and the cross-head *2*.

K is a stuffing-box in which packing is placed, and is fitted with a gland *J*, which, when bolted down, compresses the packing around the piston-rod *1*, and makes a steam-tight joint. This packing is usually made in the form of split rings, which are so placed that the split of the first ring is covered by the solid part of the next ring. When repacking, care should be taken not to cause unnecessary friction by too much pressure from the gland. The cross-head *2* is given an easy sliding fit between the guide-bars, which are in line with the path of the piston-rod, and combine with the cross-head to relieve the piston-rod of all bending strains.

2040. The connecting-rod *4* forms the connecting-link between the cross-head and crank *5*. The joint between the cross-head *2* and connecting-rod *4* is made by the cross-head pin *3*, and that between the connecting-rod and crank by the crank-pin *6*. Connecting-rods are usually made from 2 to 3 times the length of the stroke, or from 4 to 6 times the length of the crank, or from 4 to 6 "cranks" in length, as it is called.

The crank-shaft *7* forms a rigid connection between the crank *5*, the eccentric *9*, and the fly-wheel *8*. The power developed in the engine is, therefore, transmitted through the shaft. When the engine is running, all the energy which has been expended in giving the fly-wheel its speed is stored up in the fly-wheel. This energy, from the law of

the conservation of energy, is utilized in assisting the steam-power in overcoming any sudden change in the load on the engine, as well as in carrying the crank over its dead-center positions. These occur twice in every revolution of the crank—when the piston reaches the end of its stroke, and the centers of the cross-head pin *3*, crank-pin *6*, and crank-shaft *7* all lie in the same straight line.

2041. The eccentric *9*, which imparts motion to the slide-valve *V*, is the exact equivalent of a crank having the same throw. This is clearly shown in Fig. 669, in which *9*
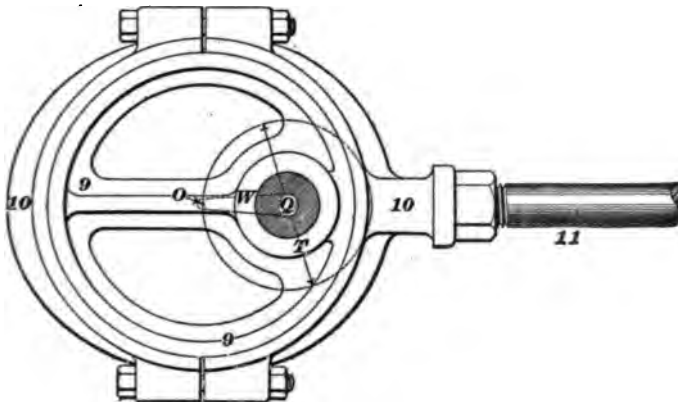
Fig. 669.

is the eccentric, *10* is its strap, *11* is the eccentric-rod, and *W* is a crank having a throw *T*, equal to that of the eccentric. The center of the crank-shaft *Q* is the center of rotary motion. The dotted circle represents the path of the common center *O* of the eccentric and crank *W*. The eccentric revolves freely in the eccentric-strap *10*, which is rigidly connected to the eccentric-rod *11*. (See Figs. 667 and 669.) In practice, the diameter of the shaft generally exceeds the throw of the eccentric. In plain slide-valve engines the eccentric is usually keyed to the shaft after being properly adjusted. The connection between the eccentric-rod *11*, Fig. 667, and the valve-rod or stem *13*

varies slightly in different engines of this class. The illustration exhibits a common form called the rocker connection.

THE PLAIN SLIDE-VALVE.

ACTION OF THE SLIDE-VALVE.

2042. In the discussions which follow, it must be remembered that the steam enters the steam-chest as it comes from the boiler through the steam-pipe S (Figs. 667 and 668), and is called live steam ; that, while it is within the cylinder of the engine, the heat stored in the steam is the active agent in the accomplishment of the results there obtained, and that all the steam discharged from the cylinder passes out of the exhaust-pipe O, Fig. 667, through the exhaust-port E, Fig. 668. It must also be understood that the motion of the *piston* is imparted to the *crank*, and that the *slide-valve* receives its motion from the *eccentric*. The crank and the eccentric have the same rotary motion, in consequence of the rigid shaft connection between them. The angle between the center lines of the crank and the eccentric is always a little more than a right angle.

2043. In order to show the action of the slide-valve, a series of skeleton diagrams, Figs. 670 to 672, have been drawn.

These diagrams have been distorted, in order that the eccentric radius might be long enough to show up well; it is three times as long as it should be for the amount of valve movement shown by the figure. The diameter of the crank circle is also a little greater than the stroke of the piston for the same reason. In order to show the distribution of steam by the valve, a diagram has been drawn above and below each cylinder, those above being marked M, and those below, N. These diagrams are supposed to be drawn in the following manner: Imagine it to be possible to connect two small pipes to the piston, one on each side. Suppose each pipe has a steam-tight piston working in it, the lower side of the pistons being subjected to the steam pressure in the cylinder, and the upper side to the atmospheric

pressure. Suppose, further, that there is a coiled spring
on top of the piston; that a piston-rod passes through
the center of the spring, and that a pencil is attached to
the end of the piston-rod. If a pressure of 10 pounds is re-
quired to compress the spring 1 inch, it is evident that for
every 10 pounds pressure in the cylinder, the pencil will
move upwards 1 inch, and, if it touched a sheet of paper,
would mark a line on that paper. It will now be presumed
that an arrangement like that just described is attached to
the steam-engine piston, and that the pencil touches a sheet
of paper, which is held stationary. Then, when the steam-
piston moves ahead, the pencil will make straight lines at
heights corresponding to the steam-pressure on the under
sides of the little pistons, except when the pressure of the
steam in cylinder varies, in which case the pencil will
move up or down, according as the pressure increases or
diminishes.

Having made these suppositions clear, let Q X, Figs. 670
to 672, represent the line which the pencil would trace if
there were a perfect vacuum in the cylinder; i. e., Q X is
the line of zero pressure, or the **vacuum line;** also let A B
represent the **atmospheric line,** or the line which the
pencil would trace if the pressure in the cylinder was just
equal to that of the atmosphere, and Q Y the line of no
volume. Then, the point Q represents no volume and no
pressure. Finally, let 1 D represent the volume of the
clearance; that is, the space between the piston and cylinder-
head when the piston is at the end of its stroke.

2044. Consider Fig. 670 (a). The piston is represented
as just beginning the forward stroke, and the valve as
just commencing to open the left steam-port, both moving
in the same direction, as shown by the arrows. If the valve
had no outside lap (see Art. **2047**), the position of the
eccentric center would be at c, but on account of the lap,
the valve has moved ahead of its central position in order to
bring its edge to the edge of the port. To accomplish this,
the eccentric center has been moved from c to b, O b being
the position of the eccentric radius. The angle b O c,

which the eccentric radius makes with the position it would
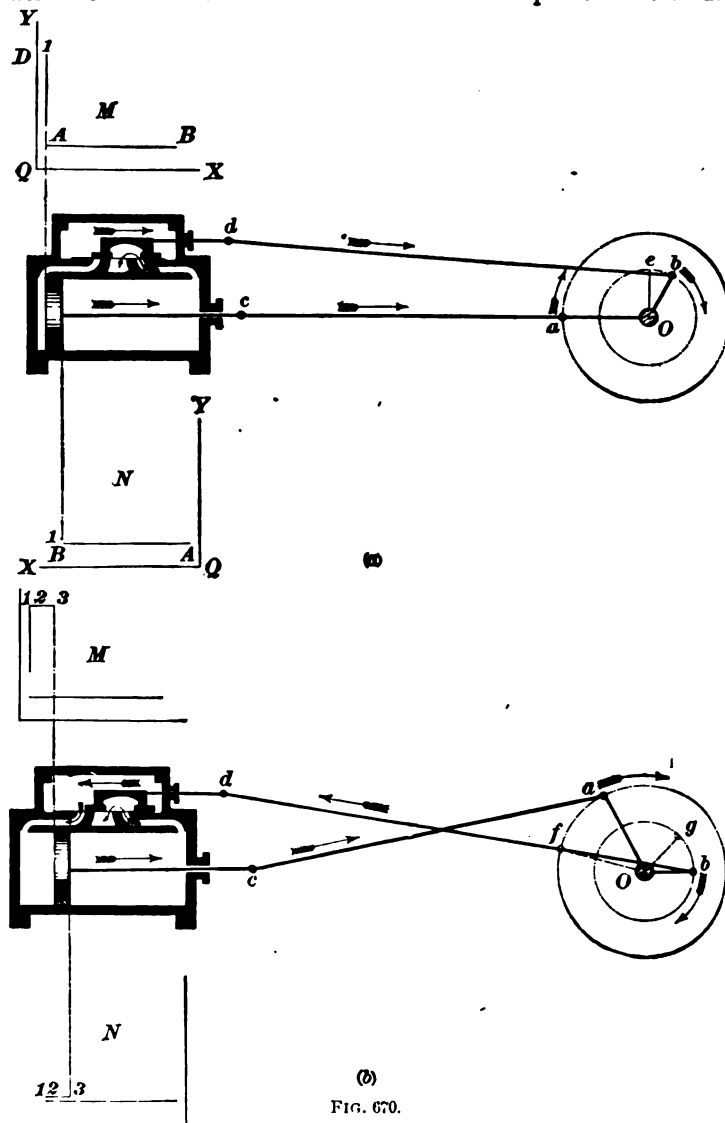
FIG. 670.

be in if there were no lap or lead, is called the **angle of advance.**

Assume that the piston and valve have moved a very small distance, just sufficient to admit steam to fill the clearance-space on the left of the piston, so that the steam acts on the piston at full boiler-pressure. If the length of the line *A 1* represents the boiler-pressure (gauge), the pencil which registers the pressure on the left side of the piston will be at *1*. The steam on the right side of the piston is flowing (*exhausting*) into the atmosphere through the exhaust-port, as shown by the arrow. As the size of the exhaust-port is limited by practical considerations, the exhaust is not perfectly free, and there is a slight pressure on the exhaust side of the piston, in addition to the atmospheric pressure. This is termed **back-pressure.** Therefore, in the diagram *N*, let *1* be the position of the second pencil; then, *1 B* is the back-pressure.

In Fig. 670 (*b*) the piston has advanced far enough to enable the valve to reach the end of its stroke and open the port its full width. The crank and eccentric have moved to the positions *O a* and *O b*. The eccentric radius is horizontal, and any further movement of the crank will cause the eccentric to travel in the lower half of its circle and make the valve move back. In the diagrams *M* and *N*, the pencil has traced the lines *1–3*.

Fig. 671 (*a*) marks one of the most important points of the stroke. Here the valve has closed the steam-port, i. e., *cut off* the steam, and from here to the end of the stroke, the steam in the cylinder expands. This point of the stroke is called the **point of cut-off.**

The exhaust-port is now partially closed. The crank and eccentric have moved to the positions indicated. During this movement, the pencils have traced the lines *3–5*.

Fig. 671 (*b*) shows another very important valve position. Here the inside edge of the valve closes the exhaust-port, and, from now on to the end of the stroke, the steam in front of the piston is compressed. This point of the stroke is called the **point of compression.** In the diagrams *M* and *N*, the lines *5–6* are traced by the pencils. The line *5–6* on the diagram *M* is an expansion line, the pressure falling

'as the piston moves ahead.   This period during which the
pressure falls is called the **period of expansion.**
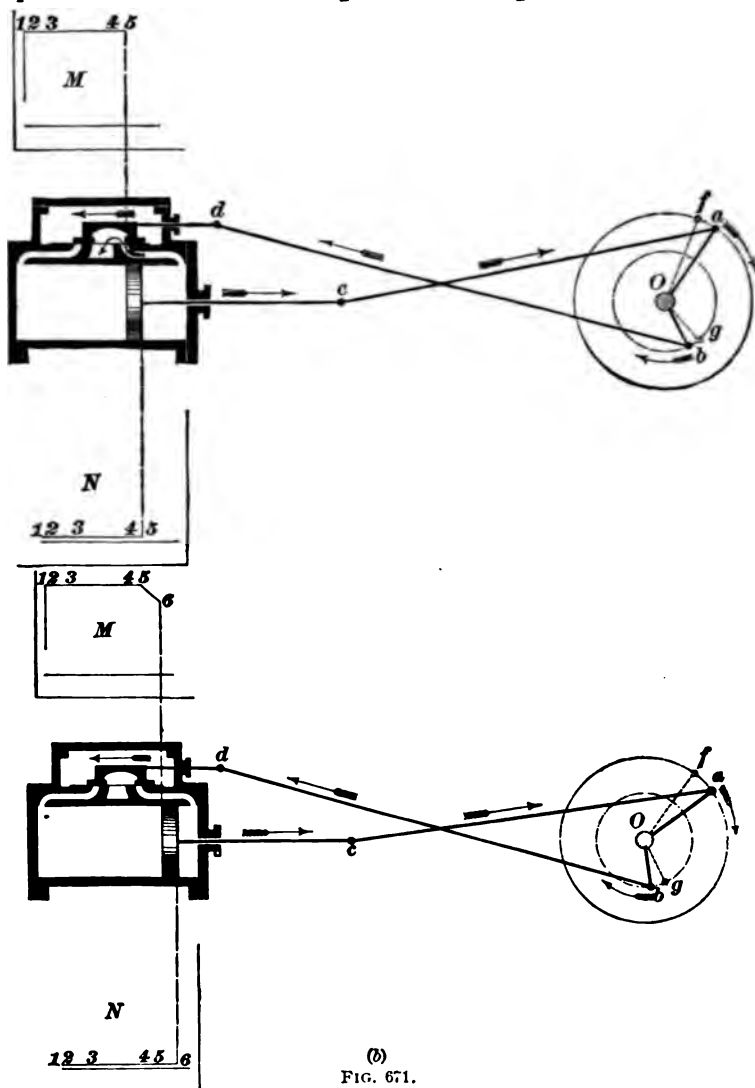
(b)
FIG. 671.

In Fig. 672 (*a*) the piston has advanced **far enough to**
cause the left inside edge of the valve to be in line with the

inside edge of the left port. The slightest movement of
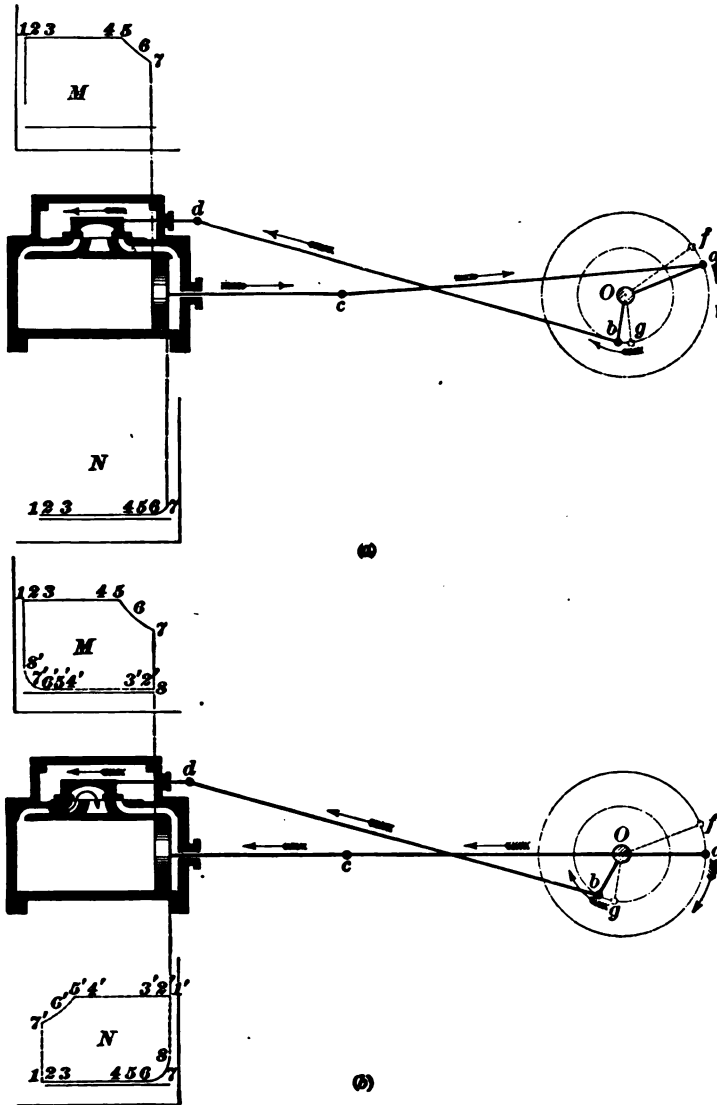
FIG. 672.

the valve to the left will open the left port to exhaust.
This point of the stroke is called the **point of release**

Expansion really ends here, although, on account of the limitation in the size of the ports, there will still be a slight further expansion, owing to the inability of the steam to escape instantly.   During this last movement of the piston, the pencils trace the lines *6–7* on the diagrams *M* and *N*. On the diagram *M*, the line *6–7* is a continuation of the expansion line *5–6*, while in the diagram *N* it shows part of the compression line, the pressure rapidly increasing as the piston nears the end of the stroke.

In Fig. 672 (*b*) the piston has reached the end of its forward stroke, and is about to begin the return stroke.   The right outside edge of the valve is in line with the outside edge of the right port.   The steam is exhausting from the head end of the cylinder, as shown by the arrows.   The crank and eccentric are both diametrically opposite their positions in Fig. 670 (*a*).   In the diagrams *M* and *N*, the pencils have traced the lines *7–8*.   *M* shows that the pressure has fallen very rapidly from *7* to *8*, while in *N* it has risen from *7* to *8*.   The very slightest movement of the piston to the left will admit steam to the crank end of the cylinder and cause the pencil to rise to the point *1'*.

During the return stroke, the above-described actions of the steam will be repeated, the pencils tracing the dotted lines on the diagrams *M* and *N* in Fig. 672 (*b*), the exhaust going through the left port and the steam through the right port.   As the process is so nearly like the preceding, the diagrams have not been drawn, but the student should follow the valve through the different positions, and note the effects on the diagrams.   To assist him in this, the corresponding points have been numbered as in the foregoing figures.

2045.   Lead.—A valve is said to have **lead** when it commences to open the steam-port just before the piston reaches the end of the stroke.   The amount of lead is measured by the distance between the edge of the valve and the edge of the port from which the valve is traveling.   In Fig. 673, the lead is the distance *a b*.   Most engineers give their valves lead in order to have the clearance-space filled with

steam at boiler-pressure when the piston begins its stroke. The effect of lead on the angular advance of the eccentric
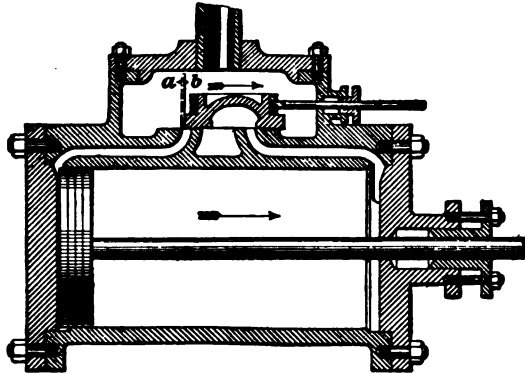
FIG. 673.

is evidently the same as an increase of lap; i. e., it increases the angular advance.

2046. In Fig. 674 is shown a sectional view of a plain slide-valve with its center n in line with the center m of the exhaust-port E. The valve takes this position during
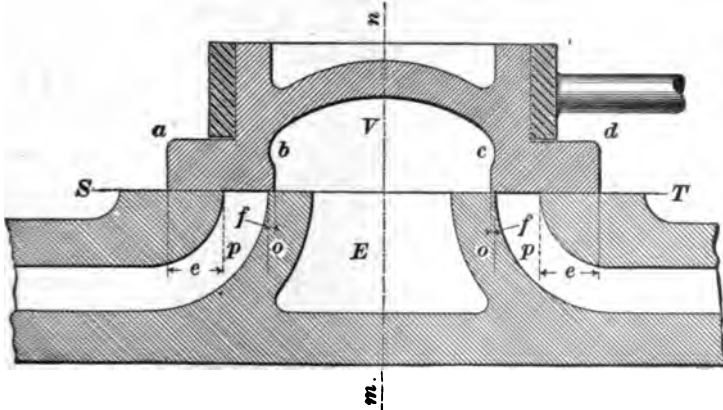
FIG. 674.

the interval between the point of release of the steam from the head end of the cylinder, and the point of compression of the steam in the crank end of the cylinder, during the forward stroke of the piston, and conversely for

the backward stroke. *S T* is the valve-seat. The flange
face *a b* or *c d* is the lip of the valve. The portion *e* of the
flange face is the **outside** or **steam lap** of the valve, while
portion *f* is the **inside lap.**

2047. Effects of Lap.—The study of Figs. 670 to
672 should show the effects caused by varying the lap.
Thus, in Fig. 671 (*a*), it is evident that if the outside lap had
been less, the valve would not close the left port when its
center was in the position shown; consequently, the piston
must move farther ahead before the valve can move back
far enough to close the port. This, of course, makes the
cut-off take place later in the stroke, and shortens the
expansion. It is likewise evident that if the valve had more
lap, this extra lap would extend beyond the port when the
center of the valve was in the position shown. Therefore,
the valve would cut off earlier in the stroke, and the expan-
sion would be lengthened. Hence, *increasing the outside lap
means an earlier cut-off and an increased expansion, while
decreasing the outside lap means a later cut-off and a dimin-
ished expansion.*

Considering the inside lap, it is evident from Fig. 671 (*b*)
that, if the inside lap had been less, the exhaust-port would
not have closed so soon, and, consequently, the compression
would have begun later; had the inside lap been greater, the
compression would have begun earlier. Fig. 672 (*a*) shows
that, with a diminished inside lap, the exhaust (usually
termed **release**) would begin earlier, while with an increased
inside lap, the release would have taken place later in the
stroke. Hence, *increasing the inside lap increases compres-
sion and delays the release, while diminishing the inside lap
decreases compression and hastens release.*

**TO SET THE PLAIN SLIDE-VALVE AND ADJUST THE
ECCENTRIC.**

2048. In order to set the valve, we must first bring the
crank to either its *head* or *crank* dead-center position.
These occur respectively as the piston completes its back-
ward and its forward strokes, when the centers of the cross-

head pin, crank-pin, and crank-shaft all lie in the same straight line.

2049. To Find the Dead Centers.—Make a fine, clear mark on the cross-head, then, while an assistant rotates the fly-wheel *in the direction opposite to that in which the engine is to run*, or backwards, follow the mark on the cross-head with a sharp instrument until it reaches successively the two extreme points of its travel, and at these points make marks on the guide-bar opposite the cross-head mark.

After this has been done for both head and crank positions of the cross-head, the engine should again be rotated *backwards* very slowly, and the accuracy of the guide-bar marks tested. We now have the dead-center points, since the centers of the cross-head pin, crank-pin, and crank-shaft will lie in the same straight line whenever the cross-head mark comes opposite either one of the guide-bar marks.

2050. To Set the Slide-Valve.—Remove the steam-chest cover by loosening the nuts with which it is screwed down, as shown in Fig. 668. Then revolve the crank *forwards*, or *in the direction in which it is to run*, until the cross-head mark coincides with the head guide-bar mark. Then if the eccentric-rod is *directly connected* to the valve-stem, that is, if the direction of motion of both the eccentric-rod and valve-stem is always the same, place the eccentric on the shaft a little more than a right angle *ahead* or in advance of the crank. But if the eccentric-rod is *cross-connected* to the valve-stem, that is, if its direction of motion is always directly opposite to the motion of the valve-stem, place the eccentric on the shaft a little more than a right angle *behind* or following the crank. Continue in either case to increase the angle between the eccentric and crank by turning the eccentric on the shaft, till the point *a* of the valve, Fig. 674, coincides with the point *p*. Then rotate the engine *forwards* till the cross-head mark coincides with the crank guide-bar mark, and see if *d*, Fig. 674, coincides with *p* of the right-hand port. If so, the valve is set correctly, but if *d* has passed *p*, the valve-stem *13*, Fig. 667,

must be removed and shortened an amount equal to on(
half the distance between p and d; and if d does not com
up to p, that is, if the right-hand port remains open, tl
valve-stem must be lengthened an amount equal to one-ha
the distance between d and p. The valve-stem now havir
the proper length, rotate the fly-wheel forwards till the cros
head mark again coincides with the head guide-bar mar
replace the valve-stem, and turn the eccentric on the sha
until the distance between p and a is equal to the desir(
lead, and firmly secure the eccentric to the shaft. Repla(
the steam-chest cover, as the slide-valve is set.

The lead usually given to a plain slide-valve is about
of an inch for quiet running, but at any time it may be i
creased or diminished without opening the steam-ches
simply by turning the eccentric very slightly forwards
backwards.

2051. Slide-valve engines are made to cut off at diffe
ent points in the stroke, according to the conditions und
which they are to be operated. That is, the point of cu
off 2, Fig. 681, may be made to occur earlier or later
the stroke than the figure represents. The point of cut-c
is expressed in the form of a ratio, or coefficient. Thus,
Fig. 681 the length of the stroke is represented by tl
length of the line $A Z$, which is equal to the length of tl
diagram, and for a stroke of 42 inches, this line may be div
ded into 42 equal divisions, making the length of eac
division represent one inch. Then, by producing the lir
$2-10$, we see that the steam is cut off when the piston h:
traveled 28 inches, or $\frac{28}{42} = \frac{2}{3}$ of its stroke.

2052. The **ratio of expansion** is the number (
times the volume of the live steam in the cylinder at tl
instant of cut-off is increased during the period of expansior
It is determined by dividing the length of the stroke b
the distance through which the piston has moved whe
cut-off occurs. In the case just given, the stroke
36 inches; cut-off occurs at 24 inches. The ratio of expar
sion is then $\frac{36}{24} = \frac{3}{2} = 1\frac{1}{2}$, that is, the steam expands to or

and a half times its volume at the point of cut-off, cr its volume is increased by the expansion in the cylinder an amount equal to one-half of what it was at cut-off.

In practice, the point of cut-off can be obtained directly from the engine. Measure the distance between the dead-center points as marked on the guide-bars; this will be the length of the stroke. Take off the steam-chest cover, and with the piston at the head end of the cylinder, slowly rotate the engine forwards until the head edge of the slide-valve coincides with the head edge of the steam-port. Then measure the distance between the head dead-center guide-bar mark and the mark on the cross-head; divide this latter quantity by the one first taken, and the result will be the cut-off required, in a fraction of the stroke. Plain slide-valves usually cut off between ½ and full stroke.

CORLISS VALVE-GEAR.

2053. As has been stated before, the plain slide-valve involves all the principles made use of in any of the more complicated forms of valve-gear at present in use. The *Corliss valve-gear* is, however, being so extensively employed in different kinds of machinery, that a short description of its working parts and principles is here given.

In Fig. 675 is shown a side elevation of this valve-gear, and in Fig. 676 a section through the cylinder and valves. It has four separate and distinct valves. Two of these, v and v', Fig. 676, connect directly with the steam-chest d and steam-pipe s, and are called steam-valves. They are rigidly connected with the cranks N and N', Fig. 675, N' being removed in order to show more clearly the disengaging link l'. The other two valves, r and r', Fig. 676, connect directly with the exhaust-chest l and the exhaust-pipe o, and are called exhaust-valves; they are rigidly connected with the cranks M and M', Fig. 675. All the valves are cylindrical in form, and extend across the cylinder above and below, respectively.

A, Fig. 675, is a disk or wrist-plate, which is made to rock

24—12

upon a stud C, by the eccentric-rod B, connecting it with an eccentric on the crank-shaft.

There are four valve-stems : E and E', which connect the wrist-plate A with the bell-cranks H and H' of the steam-valves, and F and F', which connect the wrist-plate A with the cranks M and M' of the exhaust-valves. The valve-stems can be lengthened or shortened as the case may require, and the action of any one valve may be regulated independently of the other three. As the wrist-plate A rocks backwards and
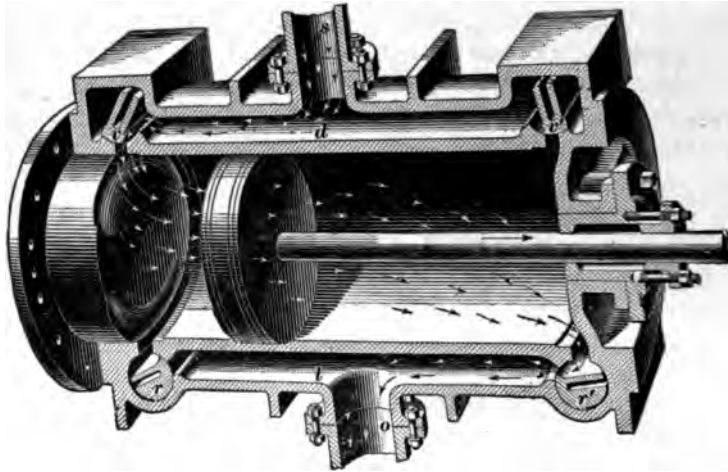
FIG. 676.

forwards, the exhaust-valves R and R', which are rigidly connected with their cranks M and M', rock with it. The bell-cranks H and H', which are provided with the disengaging links shown at I and I', are also given this rocking motion, and by hooking on to the blocks B and B', which are rigidly connected to the cranks N and N', open the steam-valves V and V'.

The projections a and a' on the two trip collars G and G' unhook these disengaging links I and I', after they have rotated the valves I^- and I'' through a certain angle, and the cranks N and N' are pulled back to their first positions by the vacuum-air dash-pots P and P', against the resistance

of which the valve-cranks N and N' were raised.   The movements of the valves open and close the steam and exhaust ports of the cylinder at the proper intervals.   The pins of the valve-stems are so located on the wrist-plate that the steam-valves V and V'' have their quickest movement while the exhaust-valves R and R' have their slowest movement, and the exhaust-valves have their quickest movement while the steam-valves have their slowest movement.   As a consequence of this arrangement, the steam and exhaust valves have entirely independent movements, and the inlet-ports may be suddenly opened full width by the quick movement of the steam-valves, while the exhaust-valves are practically motionless.   The advantage of this valve-gear is that it permits an earlier cut-off, with a greater range and a more perfect steam distribution, than is attained with the plain slide-valve.

Engines fitted with the Corliss valve-gear can not run at much more than 90 revolutions per minute.

INDICATORS AND INDICATOR-CARDS.

DESCRIPTION OF THE INDICATOR.

2054. In Fig. 681 and 682 are given diagrams (*1–2–3–4–5–6*) in which vertical distances represent pounds pressure per square inch, and horizontal distances the position of the piston in its stroke.   Such a diagram is called an **indicator-diagram.**   Indicator-diagrams are obtained by making use of an instrument called an **indicator,** Fig. 677, which is fitted to the steam-engine cylinder, as shown in Fig. 679. Holes are drilled into the clearance-spaces of the steam-cylinder (see H and C, Fig. 679), and connected with a pipe O, having a three-way cock Q in the middle.   The indicator is securely fitted to the arm F of the three-way cock by inserting the projection s (see Fig. 678) in the end of the arm, and tightening up by the nut r.

It is the office of the three-way cock to check the passage of the steam when the indicator is not in use, but it can be

so turned as to give a free passage of steam through H and F, while closing it through C, or to give a free passage through C and F, while closing it through H. Referring now to Fig. 678, which is a sectional view of the indicator, a is the cylinder of the indicator in which the piston g slides. The spring d resists any upward motion of the piston g, but when such a motion is given to the piston, it is transmitted through the piston-rod e and the link i to the point k of the lever $n\,k\,p$. These parts are so adjusted that, as the lever
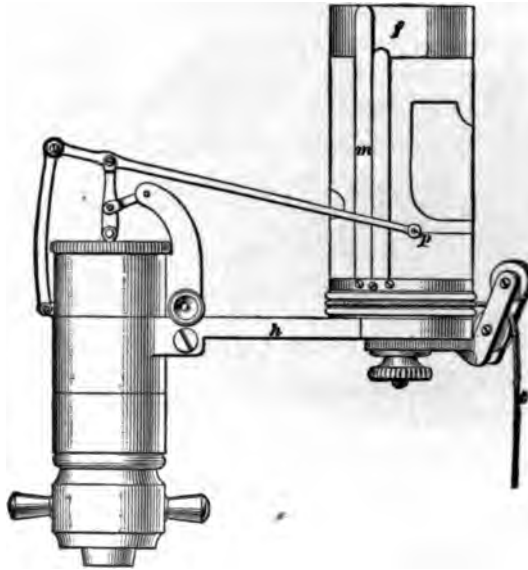
FIG. 677.

swings about n as a center, a pencil-lead at p, the extremity of the arm $n\,k\,p$, will mark a straight vertical line on the drum f. If, now, we open the pipe-connection $H\,Q\,F$, Fig. 679, to a free passage of steam from the steam-cylinder to the indicator (the passage $Q\,C$ being closed by the cock at Q), it is evident that as the steam-pressure in the steam-cylinder varies during a stroke of the engine-piston, the movement of the indicator-piston g, Fig. 678, will be effected by the changing steam-pressure, and since the spring d of the indicator is so made that it will require a definite number of

pounds pressure per square inch on the piston g to compress it sufficiently to move the pencil-lead at p vertically one inch on the drum f, therefore the pencil-lead at p will mark on the drum f vertical lines proportional to the pressure be-hind the piston, during the various points of its stroke. If we now close the passage $H\,Q$ to steam and open the passage $C\,Q\,F$, by turning the cock at Q, the pencil will in this case
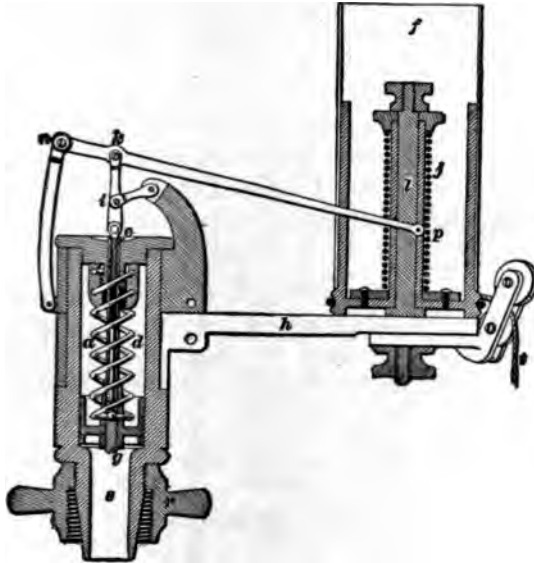
Fig. 678.

have a vertical movement proportional to the steam-pressure behind the piston in the crank end of the steam-cylinder.

2055. The **scale** of an indicator-spring is the number of pounds pressure per square inch on the indicator-piston necessary to give the pencil-lead a movement through a vertical distance of one inch, and it should not be less than ¼ the boiler-pressure under which the engine is operated.

The spindle l, Fig. 678, is firmly fixed to the bar h, and is also connected to the drum f by means of the spring j. If, now, the cord t, which is wound on the drum, is pulled, the drum f will rotate against the action of the spring, but it

will, in turn, be rotated back to its first position by the spring when the pull on the cord *t* is discontinued.

From this it is evident that, if we pull and release the
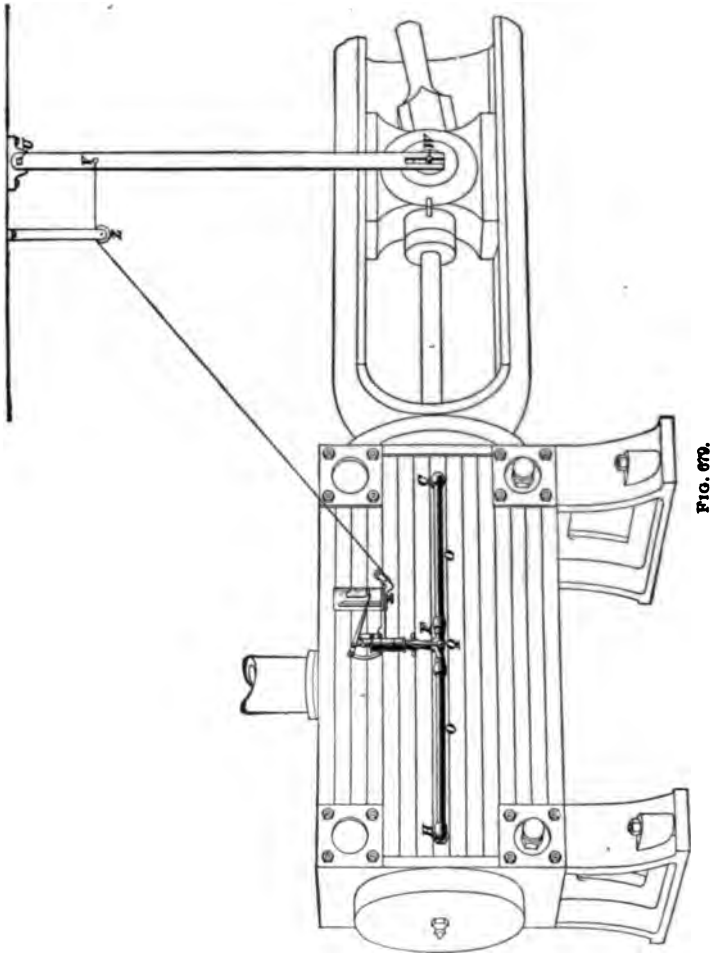
FIG. 678.

string by a forward and backward motion of the hand, a corresponding rotary motion will be given to the drum. When the indicator is in use, such a motion is thus given by means of the lever *U V W* and the pulley *Z*, Fig. 679.

This arrangement is termed a **reducing motion,** because it enables us to take an indicator-card which will be less in length than the stroke of the engine. The lever is held in position at the ceiling by a pin U, and its lower end, which is slotted, is made to follow the motion of the cross-head by means of the pin W, clearly shown in the figure. The pin U should be directly over the pin W when the cross-head is in the center of its stroke. The *effective length* of the lever $U V W$ is the distance $U W$ taken when the cross-head is in the center of its stroke.

2056. To determine at what point the cord of the in-dicator-drum is to be fastened to the lever, in order to give the drum a rotary movement through a distance equal to the length of the indicator-card, when the effective length of the lever, the length of the stroke of the piston, and the length of the card are given:

Rule.—*Multiply the effective length of the lever in inches by the length of the card in inches, and divide this product by the length of the stroke in inches; the quotient is the dis-tance in inches below the center of the pin U at which the cord is to be attached to the lever.*

EXAMPLE.—For the engine shown in Fig. 679, the stroke is 36 inches; the effective length of the lever is 72 inches, and the length of card desired is 3.5 inches; what is the distance of the point on the lever below the center of the fulcrum, at which the cord is to be attached?

SOLUTION.— $\frac{72 \times 3.5}{36} = 7$ in. Ans.

If, when the crank is on the *head* dead-center, we pass the cord over the pulley Z, Fig. 679, and fasten it at V, after drawing it just tight enough to rotate the drum through about half an inch, it is evident that when the engine is running, the drum will be given a rotary motion, backwards and forwards, corresponding to the stroke of the engine. The cord should only be fastened to the reducing motion during the time in which a card is being taken. It must also be remembered that the length of the indicator-diagram has nothing whatever to do with the results obtained from

it, but is made less than the stroke because the drum will not rotate so great a distance.

DIRECTIONS FOR TAKING INDICATOR-DIAGRAMS.

2057. Having everything arranged as explained, and the engine running under its usual load, take a piece of paper about seven inches long and three and one-half inches wide, and fasten it around the drum by passing the ends under the card-holders *m*, Fig. 677. Now, connect the cord *t* to the reducing motion, and let the pencil-lead at *p* press lightly against the drum as it rotates. A straight line will be made on the paper. This is the atmospheric line *A Z*,
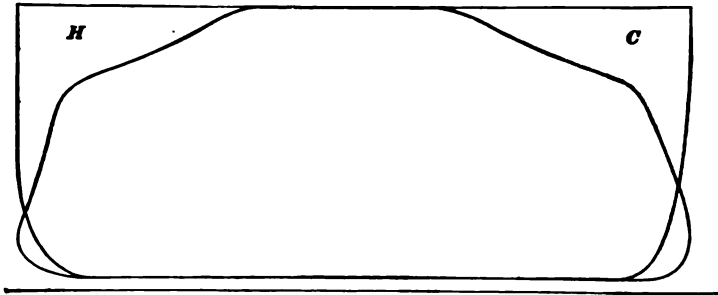
FIG. 680.

Fig. 681. Next, turn the cock at *Q* so as to admit steam to the indicator from *H*, Fig. 679, the head end of the cylinder, and the pencil-lead will move up and down as the piston of the engine goes forwards and back.

Turn the cock again and admit steam to the indicator from the crank end of the steam-cylinder, and the pencil-lead will again move up and down, having in each case drawn such a figure on the paper as is shown in Fig. 677 on the drum. In Fig. 680 are shown the head and crank cards, *H* and *C*, as they were drawn by the pencil during the two strokes of the piston.

Before removing the indicator-card from the drum, shut off the steam and disconnect the drum-cord from the reducing motion.

2058. In Fig. 680, *H* is the diagram from the head end of the steam-cylinder, and *C* is the diagram from the crank end.

If desired, these may be taken on separate pieces of paper, as shown in Figs. 681 and 682. The actual diagrams shown
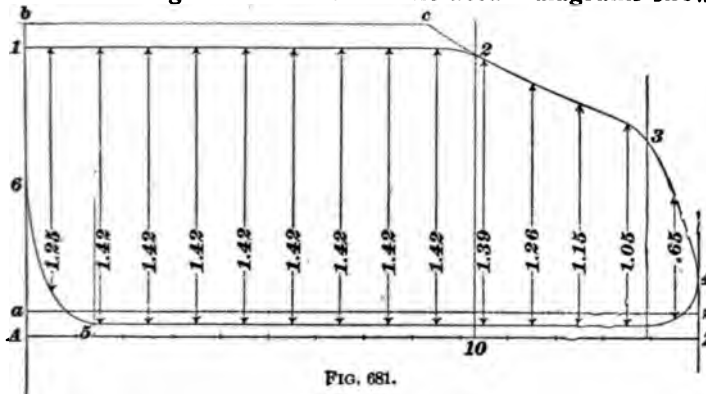
FIG. 681.

in Fig. 680 are not so regular as the theoretical ones in Fig. 672 (*b*). The different points of the stroke are, however, quite clearly defined.

FIG. 682.
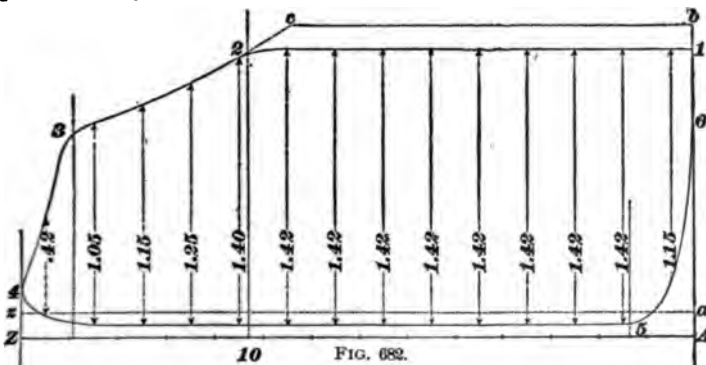
Thus: *1* is the beginning of the stroke.
       2 is the point of cut-off.
       3 is the point of release.
       4 is the end of the stroke.
       5 is the point of compression.
       6 is the point of admission.

6–1 is the admission line.

1–2 is the steam line.

2–3 is the expansion curve.

3–4–5 is the period of release.

4–5 is the back-pressure line.

5–6 is the compression curve.

A Z is the atmospheric line.

2059. To determine from an indicator-card, taken from the cylinder of an engine, at what point in the stroke of the engine *cut-off* occurs:

Rule.—*Measure along the atmospheric line the distance in inches between the extreme projections of the card. Measure also the distance in inches between the projections of the points of admission and cut-off on the atmospheric line, and divide the latter quantity by the former.*

EXAMPLE.—In Fig. 681, the distance between *A* and *Z* is 3.5 inches. The distance between *A* and *10* is 2.33 inches; when did the valve cut off ?

SOLUTION.— $\dfrac{2.33}{3.5} = .666 = \frac{2}{3}$; therefore, the cut-off occurs at $\frac{2}{3}$ stroke. Ans.

INDICATED HORSEPOWER.

2060. The **horsepower** developed by the engine may be found directly from the area of the indicator-diagram. It is more convenient, however, to use the diagram to find the "mean effective pressure" exerted on the piston.

2061. The **mean effective pressure,** or M. E. P., is defined as the average net pressure urging the piston forwards during its entire stroke in one direction. The mean effective pressure may be found in two ways:

2062. The area of the diagram in square inches may be found by an instrument called the planimeter; *the M.E.P. is then found by dividing the area of the diagram in square inches by the length of the diagram in inches, and multiplying by the scale of the spring.*

EXAMPLE.—The area of the diagram is 4.2 sq. in., and the length is 3.5 in.; a 40 spring being used, find the M. E. P.

SOLUTION.— $\dfrac{4.2}{3.5} \times 40 = 48$ lb. per sq. in., M. E. P. Ans.

2063. Where a planimeter is not available, the following method of finding the M. E. P. is fairly rapid and accurate:

Draw tangents to each end of the diagram perpendicular to the atmospheric line. Divide the horizontal distance between the tangents into 10 or more equal parts. (10 or 20 parts are the most convenient, but any other number may be used.) Indicate by a dot on the diagram the center of each division, and draw lines through these dots, parallel to the tangents, from the upper line to the lower line of the diagram. On a strip of paper mark off successively the lengths of these lines, the total length thus representing the sum of all the lines. Divide this total length by the number of lines used, and multiply the quotient by the scale of the spring. The result will be the M. E. P.

EXAMPLE.—The projection of the diagram shown in Fig. 682 upon the atmospheric line is $A\,Z$; that is, lines perpendicular to this line, drawn through the extreme ends *1* and *4* of the diagram, cut it (the atmospheric line) in A and Z. $A\,Z$ is divided, in this case, into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself, and the sum of these lengths is 18.11 inches. The scale of the spring used in obtaining the diagram was 40 pounds; therefore, $\dfrac{18.11}{14} \times 40 = 51.74$ pounds per square inch $=$ the M. E. P. of the bottom-end diagram.

EXAMPLE.—The projection of the diagram, Fig. 681, upon the atmospheric line is the distance $A\,Z$, and it is divided, in this case, into 14 equal spaces. The length of each of the perpendicular lines drawn through the diagram opposite the centers of these spaces is marked on the line itself, and the sum of these lengths is 17.78 inches. The scale of the spring is 40 pounds; therefore, $\dfrac{17.78}{14} \times 40 = 50.8$ pounds per square inch $=$ the M. E. P. of the top-end diagram.

Therefore, the M. E. P. in the cylinder during a complete revolution of the crank is $\dfrac{51.74 + 50.8}{2} = 51.27$ pounds per square inch.

2064. The reason for dividing the diagram into 10 parts instead of some other number is that it shortens the work of calculation. Thus, in the two examples just given, if the number of divisions had been 10 instead of 14, and the sum of the ordinates had been 12.94 inches, the mean ordinate would have been $\frac{12.94}{10} = 1.294$ inches, and the M. E. P., 1.294 × 40 = 51.76 lb. per sq. in. All that is necessary is to add the ordinates and shift the decimal point one place to the left to obtain the mean ordinate when the diagram is divided into 10 equal parts. This method saves the time required to divide by some inconvenient number, such as 14.

2065. In Figs. 681 and 682, the vertical line *a b* represents the boiler-pressure, and, therefore, the dotted line *b c* is the line that the indicator-pencil would trace if the full boiler-pressure were maintained until point of cut-off. The line *b c* is not drawn by the indicator as ordinarily used; it has been added for sake of illustration.

2066. We have now all the material required for finding the work done in the engine-cylinder expressed in horsepower units.

Work is the product of force into the distance through which it moves. In the case of the engine-cylinder, the total force is the M. E. P. per square inch multiplied by the area of the piston; and the distance moved through in one minute is the number of strokes per minute multiplied by the length of the stroke.

2067. Rule.—*To find the* **indicated horsepower** *developed by the engine, multiply together the M. E. P. per square inch, the area of the piston, the length of stroke, and the number of strokes per minute. This gives the work per minute in foot-pounds. Divide the product by 33,000; the result will be the indicated horsepower of the engine.*

Let I. H. P. = indicated horsepower of engine;
    $P =$ M. E. P. in pounds per square inch;
    $A =$ area of piston in square inches;
    $L =$ length of stroke in feet;
    $N =$ number of strokes per minute.

Then, the above rule may be expressed thus:

$$\text{I. H. P.} = \frac{PLAN}{33,000}. \qquad \textbf{(143.)}$$

2068. The number of strokes per minute is twice the number of revolutions per minute. For example, if an engine runs at a speed of 210 revolutions per minute, it makes 420 strokes per minute. A few types of engines, however, are single-acting; that is, the steam acts on only one side of the piston. Such are the Westinghouse, the Willans, and others. In this case, only one stroke per revolution does work, and, consequently, the number of strokes per minute to be used in the above rule is the same as the number of revolutions per minute. As most steam-engines are double-acting, no mention is generally made of this fact. When the dimensions of an engine are given, unless it is stated that the engine is single-acting, it may be assumed that a double-acting engine is meant and that work is done during each stroke.

EXAMPLE.—The diameter of the piston of an engine is 10 inches, and the length of stroke 15 inches. It makes 250 revolutions per minute, with a M. E. P. of 40 pounds per square inch. What is the horsepower ?

SOLUTION.—As it is not stated whether the engine is single or double acting, assume that it is double-acting. Then, the number of strokes is $250 \times 2 = 500$ per minute. Applying formula **143**,

$$\text{I. H. P.} = \frac{PLAN}{33,000} = \frac{40 \times \tfrac{15}{12} \times (10^2 \times .7854) \times 500}{33,000} = 59.5 \text{ H. P.}$$

2069. Approximate Determination of M. E. P.— To approximately determine the M. E. P. of an engine, when the point of apparent cut-off is known and the boiler-pressure, or the pressure per square inch in the boiler from which the supply of steam is obtained, is given:

Rule.—*Add 14.7 to the gauge-pressure, and multiply the result by the number opposite the fraction indicating the point of cut-off in Table 44. Subtract 17 from the product and multiply by .9. The result is the M. E. P. for good, simple non-condensing engines.*

Or, letting p = gauge-pressure;

k = a constant (see Table 44);

M. E. P. = mean effective pressure.

Then,   M. E. P. = $.9\,[k\,(p + 14.7) - 17]$.   (144.)

TABLE 44.

Cut-off.	Constant.	Cut-off.	Constant.	Cut-off.	Constant.
$\frac{1}{8}$	.566	$\frac{3}{8}$	.771	$\frac{2}{3}$	.917
$\frac{1}{6}$	.603	.4	.789	.7	.926
$\frac{1}{4}$	.659	$\frac{1}{2}$	.847	$\frac{3}{4}$	.937
.3	.708	.6	.895	.8	.944
$\frac{1}{3}$	.743	$\frac{5}{8}$	.904	$\frac{7}{8}$	.951

2070. If the engine is a simple condensing engine, subtract the pressure in the condenser instead of 17. The fraction indicating the point of cut-off is obtained by dividing the distance that the piston has traveled when the steam is cut off by the whole length of the stroke. For a $\frac{2}{3}$ cut-off, and 92 pounds gauge-pressure in the boiler, the M. E. P. is, by the formula just given, $.9[.917(92 + 14.7) - 17] = 72.6$ lb. per sq. in.

EXAMPLE.—Find the approximate I. H. P. of a $9'' \times 12''$ non-condensing engine cutting off at $\frac{1}{2}$ stroke, and making 240 revolutions per minute. The boiler-pressure is 80 pounds, gauge.

SOLUTION.— $80 + 14.7 = 94.7$. Using formula **144** and Table 44, the constant for $\frac{1}{2}$ cut-off is .847, and $.847 \times$ boiler-pressure $= .847 \times 94.7 = 80.21$. M. E. P. $= (80.21 - 17) \times .9 = 56.89$ lb. per sq. in. Then, from formula **143**,

I. H. P. $= \dfrac{P\,L\,A\,N}{33,000} = \dfrac{56.89 \times 1\frac{2}{3} \times (.7854 \times 9^2) \times 240 \times 2}{33,000} = 52.64$ H. P.
Ans.

2071. Piston Speed.—The product LN of formula **143** gives the total distance traveled by the piston in one minute. This is called the **piston speed.** It is usual to take the stroke in inches. Then, to find the piston speed, multiply the stroke in inches by the number of strokes, and divide by 12, or, letting S represent the piston speed, $S = \dfrac{LN}{12}$, where

l is the stroke in inches. But $N = 2R$ where R represents the number of revolutions per minute. Hence,

$$S = \frac{lN}{12} = \frac{l \times 2R}{12} = \frac{lR}{6}. \quad \textbf{(145.)}$$

Rule.—*To find the piston speed of an engine, multiply the stroke in inches by the number of revolutions per minute, and divide the product by 6.*

EXAMPLE.—An engine with 52-inch stroke runs at a speed of 66 revolutions per minute. What is the piston speed?

SOLUTION.—By formula **145**, $S = \frac{lR}{6} = \frac{52 \times 66}{6} = 572$ ft. per min. Ans.

The piston speeds used in modern practice are about as follows:

	Ft. per min.
Small stationary engines	250 to 600
Large stationary engines	500 to 900
Corliss engines	400 to 750

2072. Friction Horsepower: Net Horsepower.—Formula **143** gives the indicated horsepower, or I. H. P.; that is, the total horsepower developed in the engine-cylinder. A part of the I. H. P. is used in overcoming the friction of the moving parts of the engine. The remainder is available for doing the required work.

2073. The power absorbed by the engine itself is termed the **friction horsepower.**

2074. The power available for doing useful work is termed the **net,** or **actual, horsepower.**

The actual horsepower of any engine is found by first computing its I. H. P. from a set of indicator-diagrams taken while the engine is running under full load, and then subtracting from this the I. H. P. computed from a set of indicator-diagrams taken when the engine is running under no load, but making the same number of revolutions per minute as above. The horsepower developed by the engine in this last case will only be sufficient to keep the working parts of the engine in motion at the same speed.

For example: indicator-diagrams, taken from an engine while running under full load, and having a piston speed of 498 feet per minute, show an *indicated horsepower* of 242.7. With the same piston speed, and running under no load, the indicator-diagrams show an *indicated horsepower* of 75.2. Then, 242.7 − 75.2 = 167.5 = the *actual horsepower* of the engine.

2075. The **mechanical efficiency** of an engine is the ratio of the *actual horsepower* to the *indicated horsepower;* or it is the per cent. of the mechanical energy developed in the cylinder which is utilized in the doing of useful work.

2076. To find the efficiency of an engine, when the *indicated* and *actual horsepowers* are known:

Rule.—*Divide the actual horsepower by the indicated horsepower.*

Let N. H. P. = the net, or actual, horsepower;

    I. H. P. = the indicated horsepower;

      E_m = efficiency of engine.

Then, $E_m = \dfrac{\text{N. H. P.}}{\text{I. H. P.}}.$ **(146.)**

EXAMPLE.—The indicator-diagrams taken from an engine running under full load show the I. H. P. to be 238.5. The diagrams taken when the engine is running under no load show a horsepower of 39.7. (*a*) What is the net H. P. developed by the engine? (*b*) What is the efficiency of the engine?

SOLUTION.—(*a*) Net H. P. = I. H. P. − friction H. P. = 238.5 − 39.7 = 198.8. Ans.

(*b*) By formula **146**, the efficiency is

$$\frac{\text{N. H. P.}}{\text{I. H. P.}} = \frac{198.8}{238.5} = 83.4\%.\quad \text{Ans.}$$

The mechanical efficiency of a good engine may be from 75 to 90 per cent.

The efficiency of steam-engines varies greatly; it is, however, usually taken at 66 per cent. in all approximate determinations. That is, ordinary practice shows that with the types of engine commonly used only about .66 or ⅔ of the power developed in the cylinder is actually available.

2077. We will now consider an example, to show how the above rules may be used in practical work.

EXAMPLE.—Determine approximately the dimensions of a single-cylinder, non-condensing engine to furnish 175 actual horsepower.

SOLUTION.—The I. H. P. of the engine will be about ½ greater than the actual horsepower, or $175 \times \frac{1}{2} = 262$.

This is a large stationary engine; therefore, we should have a piston speed of between 500 and 900 feet per minute, say 600 feet per minute.

The cut-off may be taken at ¼ to insure good results, and the boiler-pressure may be assumed to be 80 pounds per square inch. From formula **144,** the M. E. P. is $.9 \times [.904(80 + 14.7) - 17] = 61.75$ pounds per square inch.

Letting d = diameter of cylinder,

$$\text{I. H. P.} = \frac{d^2 \times .7854 \times 61.75 \times 600}{33,000} = 262;$$

or, $$d = \sqrt{\frac{262 \times 33,000}{.7854 \times 61.75 \times 600}} = 17.24 \text{ inches, say 17 inches.}$$

Taking the ratio of stroke to diameter of cylinder as 1.5, we have stroke $= 17 \times 1.5 = 25.5$, say 26 inches.

The number of revolutions of the crank would then be $\dfrac{600 \times 6}{26} =$ 138.5 revolutions per minute.

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EXAMPLES FOR PRACTICE.

1. The mean effective pressures of two diagrams taken from the two ends of the cylinder of an 18″ × 20″ non-condensing engine, running at 200 R. P. M. (revolutions per minute), are, respectively, 57.6 lb. per sq. in. and 60.8 lb. per sq. in. long. What is the horsepower of the engine ?   Ans. 304.335 H. P.

2. The area of an indicator-diagram, as found by the planimeter, is 2.76 square inches. The length of the diagram is 2.4 inches, and the scale of the spring is 30. What is the M. E. P. ?
                      Ans. 34½ lb. per sq. in.

3. The indicator-diagrams from an engine show a M. E. P. of 27.3 pounds per square inch. The engine has a 26″ × 48″ cylinder, and makes 68 revolutions per minute. Calculate the I. H. P. developed by the engine.   Ans. 238.94 H. P.

4. Find the I. H. P. developed by an 8″ × 12″ engine, running at 260 revolutions per minute, the average M. E. P. being 32.61 pounds per square inch.   Ans. 25.83 H. P.

5. (a) What is the piston speed of the engine of example 3 ? (b) of the engine of example 4 ?   Ans. { (a) 544 ft.
                                                          { (b) 520 ft.

CONDENSERS.

2078. In Fig. 683 is shown a condenser, of which Fig. 684 is a sectional view. The operation of this condenser may be explained as follows: When steam is admitted to the steam-cylinder, it causes the piston P to move first to the right and then to the left, as in the cylinder of a steam-engine, and since the air-pump piston O and the water-pump piston Q are both rigidly connected to P by the double piston-rod as shown, they are given the same motion
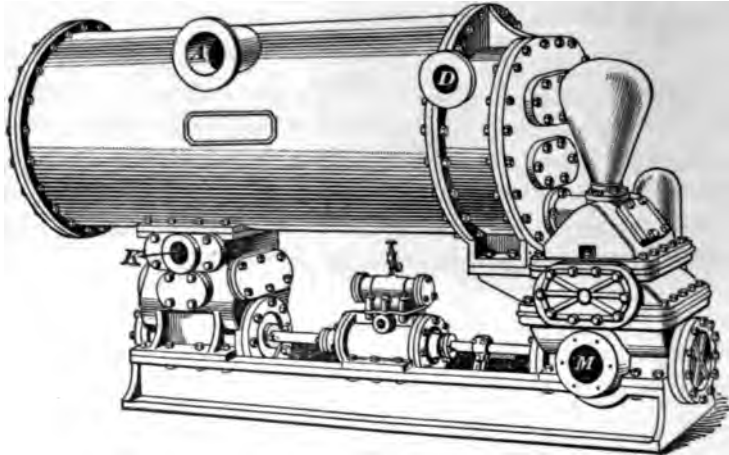
FIG. 683.

as P, with the following effect: The nozzle at M is connected by piping to a water-supply, through which water is drawn into and discharged from the circulating or water-pump cylinder by the movement of the piston Q, in the manner clearly shown by the arrows, Fig. 684. The valves S, S and V, V are automatically opened and closed by the pressure of the water below, and by the pressure of the water and springs above them. After the water is forced through the inlet C into the chamber F, it flows, as is indicated by the arrows, through the inner tubes of the lower layer of double tubing to the left, and having passed through their entire length, it returns through the space between

the outside of the inner and inside of the outer tubes into
the chamber G.   Fig. 685 shows more clearly the arrange-
ment of this "double tubing."   From G, Fig. 684, it passes
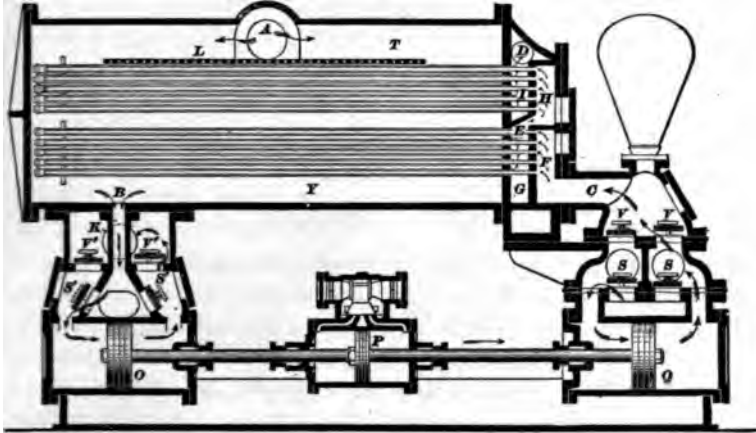
FIG. 684.

through E to H, and from H to I through the upper layer
of double tubing, as has already been explained.   From I
it is discharged through the nozzle D, carrying with it all
the heat it has received by coming in contact with the two
layers of double tubing.

The nozzle at A is connected with the exhaust-pipe of
the steam-cylinder of an engine.   The movement of the
air-pump piston O draws air through the orifice B from the
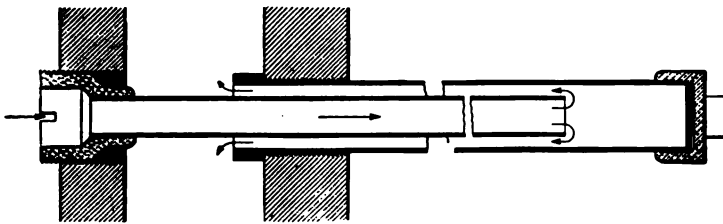
FIG. 685.

condenser-cylinder, and discharges it through the valves and
the nozzle K, in a manner clearly indicated by the arrows.
The valves S', S' and V', V' are opened and closed automat-
ically by the pressure of the air beneath them, and by the

pressure of the air and springs above them. A partial vacuum is created in the condenser-cylinder Y by the action of the air-pump; this reduces the back pressure of the steam on the engine piston, and permits the exhaust steam to leave the engine-cylinder with much less resistance than it would encounter if it were discharged against the pressure of the atmosphere.

As the exhaust steam enters the condenser-cylinder through A, it strikes a scattering plate L, which distributes it among the tubes, and protects the upper rows of tubing from the cutting effect of a direct current of steam. By coming in contact with the cold tubes the steam is condensed and falls to the bottom of the condenser-cylinder as water; this water flows through B to the air-pump cylinder, from which it is discharged, and is then pumped into the boiler before cooling. By this means a supply of boiler feed-water is obtained at a temperature nearly as high as that of the condenser.

2079. The most important use of the condenser, however, is by means of the partial vacuum produced by the condensation of the steam to relieve the exhaust side of the piston from the pressure of the atmosphere. Without a condenser the exhaust must be forced out of the cylinder against the pressure of the outside air, about 14.7 pounds per square inch, and this pressure must be overcome by the pressure of the steam on the other side of the piston. With a good condenser the pressure against which the exhaust must leave the cylinder is reduced to not more than 3 or 4 pounds per square inch. There is thus an increase in the effective pressure which can be obtained with a given boiler pressure. This results in an increase in the power which the engine can develop, together with a reduction in the steam required to do a given amount of work. To illustrate, consider the cards shown in Figs. 681 and 682. If the pressure in the condenser-cylinder were 12 pounds below the pressure of the atmosphere, it is evident that the initial pressure need have been only $60 - 12 = 48$ pounds per square inch, in order to

have produced the same cards. The atmospheric line drawn by the indicator pencil would have been at $a\ z$ at a scale distance of 12 pounds above the old atmospheric line $A\ Z$, or 6 pounds above the back pressure line 4–5. In effect every point on the card would be lowered a scale distance of 12 pounds.

STEAM-ENGINE GOVERNORS.

2080. Steam - engine governors are mechanical devices which automatically regulate the steam-supply of an engine, só that when the load on the engine is increased or decreased, or when the steam-pressure under which it operates changes, the speed of the engine will remain constant. It must not, however, be thought that the duty of the governor is to adjust the working conditions of an engine to any sudden variation of steam-pressure or load that may occur during the time of a single stroke of the piston. It is the office of the fly-wheel to respond to these rapidly changing conditions, and by the resistance which it offers to any rapid change in its velocity, to gradually absorb this sudden force in increasing and decreasing the number of its revolutions per minute. When the engine is not supplied with a fly-wheel, there are other rotating parts, such as the drum of a hoisting-engine, which serves the same purpose. Any variation of the speed of the fly-wheel is, however, met by the action of the governor, which increases or decreases the steam-supply, and thereby restricts the velocity of the fly-wheel within certain limits. The principle that insures the action of all steam-engine governors is that of the equalization of two opposing forces, which will occur only when the engine is running at its proper speed. Any variation of the speed tends to give one of these forces an increase over the other, which is expended in moving some mechanism for the adjustment of the steam-supply.

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2081. Steam-engine governors may be divided into two classes: (1) **Throttling governors,** which throttle the steam in the supply-pipe, and (2) **automatic** or **adjustable cut-off governors,** which regulate the steam-supply by changing the point of cut-off of the valve.

2082. Throttling governors are usually of the pendulum or fly-ball type. One of these is quite clearly shown at *k, m, m', n, o,* Fig. 667. When the engine is running, a rotary motion is given to the pulley *n* by means of the belt which runs over a pulley rigidly fastened to the crank-shaft. This motion is transmitted through the bevel-gears seen at *o* to the spindle *k*, which is secured to the fly-balls *m* and *m'*.

2083. Suppose the engine to be running at its proper speed; a balance will then exist between the gravity force and the centrifugal force due to the rotary motion, both of which are acting on the balls. If, now, from any cause, the speed lessens, the centrifugal force will diminish, and gravity, acting on the balls, will pull them down. This movement on the part of the balls will, in turn, be imparted to a balanced throttle-valve, which will be opened wider, causing an increase in the initial pressure of the live steam. The live steam will now, in consequence of the additional amount of work it is capable of doing, exert more energy on the piston and bring the speed back to its proper point. If, on the other hand, the speed be increased, the fly-balls will rise upwards in consequence of the centrifugal force becoming greater than the attraction of gravity, and the steam orifice of the throttle-valve will be diminished in area. This will lower the initial pressure; the steam will consequently exert a less effort, and the speed will drop to its proper point.

2084. As an example of the automatic adjustable cut-off governor, of which there are many forms, we will consider that usually employed on the Corliss type of engines, the valve-gear of which has already been described in Art. **2053.**

Referring to Fig. 686, we see that a rotary motion is imparted to the fly-balls *m, m,* by means of a belt *p*, pulleys

s and *r*, and bevel-gear connection *o*, similar to that already described when stating the principle of the throttling governor.

Suppose that the engine is running at its proper speed. The fly-balls will then be held in their normal position by the balance existing between the centrifugal and gravity forces acting on the fly-balls *m, m*.   Suppose, now, the speed of the engine increases from any cause whatever; the centrifugal force acting on the fly-balls will also increase and will continue to pull them out, that is, to increase the
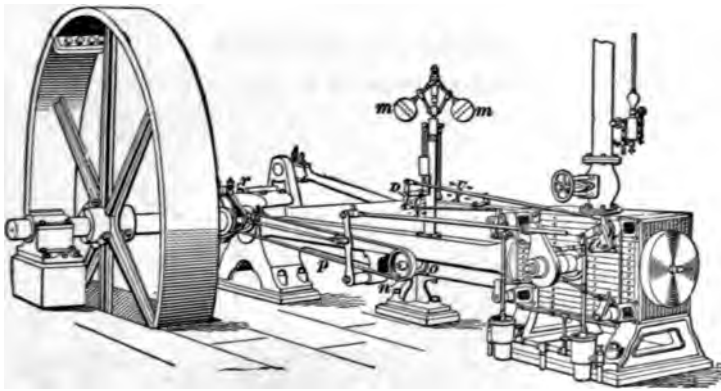
FIG. 686.

diameter of the circle in which they rotate, until a new balance is effected between it and the attraction of gravity. This movement of the fly-balls will be transmitted to the lever *D*, causing it to turn slightly about its center in the direction of the arrow *X*.   The movement of *D* will cause the trip-collars *G* and *G'*, Fig. 675, to turn through a small angle in such a direction that their projections *a* and *a'* will unhook the disengaging links *I* and *I'* earlier in the stroke of the engine.   This will cause the point of cut-off to occur earlier in the stroke, and a decrease in the speed of the engine, on account of the reduction in the amount of steam admitted to the cylinder and an increased ratio of expansion of the steam under the same initial pressure.   Should the speed from any cause diminish, a reverse operation would

be the result of the action of the governor.   The fly-balls
would drop slightly; *D* would turn as indicated by the arrow
U, and the trip collars *G* and *G'* would be rotated in such a
manner as to cause their projections *a* and *a'* to unhook the
disengaging links *I* and *I'* later in the stroke; the cut-off
would then occur later in the stroke, and a diminished ratio
of expansion at the same pressure would bring the speed up
to its proper point again.

SPECIAL TYPES OF ENGINES.

HOISTING-ENGINES.

2085.   A **hoisting-engine** is usually a combination of
two single-cylinder engines of exactly the same description
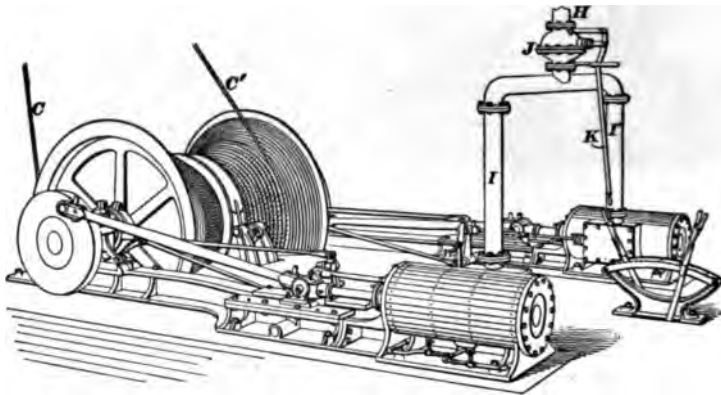
FIG. 687.

and dimensions, which have their cranks rigidly connected
to a common crank-shaft, and take steam at the same pres-
sure.   Such a combination is called a **duplex engine.**   In
order to prevent the possibility of both cranks being on a
dead center at the same moment, one crank is placed a dis-
tance of one right angle in advance of the other.   In Fig.
687 is shown such an engine.   The large double spiral drum
performs the duty of a fly-wheel, while also doing duty as
the drum on which the ropes *C* and *C'* are wound and un-
wound.   It is evident that the engine must be so constructed
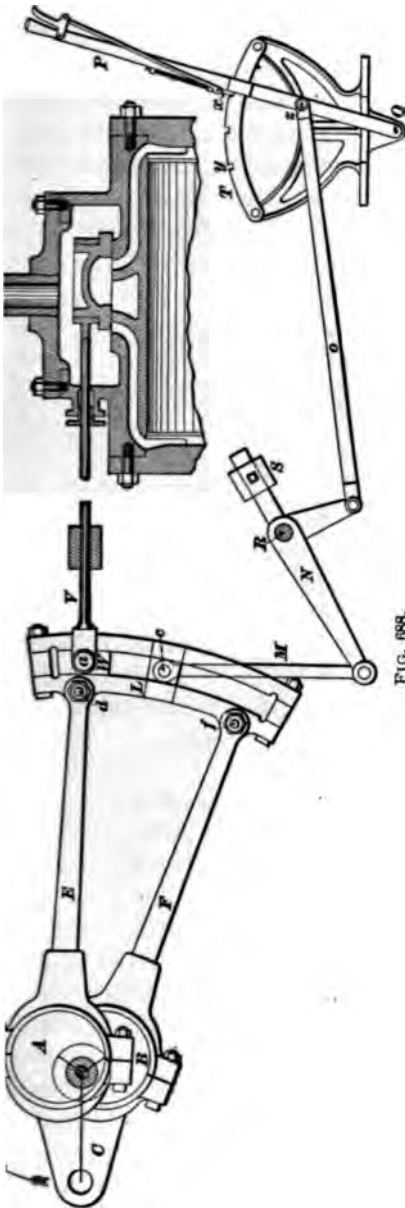
FIG. 688.

that the direction of rotation of the drum may be reversed at will. This introduces a new feature, namely, a **reversing-gear,** which must form a part of every engine of which the direction of motion is to be reversed. The most common form of reversing-gear is the Stephenson link-motion, which can be partly seen in Fig. 687, but is more clearly shown in Fig. 688; the lettering, however, applies alike to both figures.

2086. Let O be the center of rotation of the crank C, and suppose the arrow to represent the forward rotation of the engine. Then, A will be the forward eccentric and B the backward eccentric; E will be the forward eccentric-rod, and F the backward eccentric-rod.

The forward eccentric, for reasons already explained, must be slightly more than a right angle in advance

of the crank when it is directly connected, since it is to supply the means of operating the slide-valve when the engine is rotating forwards. The parts as shown are in the positions required to rotate the engine forwards. For the same reason, the backward eccentric B must be slightly more than a right angle behind the crank, when the engine is rotating forwards, so that when the engine is reversed, and f takes the place of d, B may be in advance of the crank when the engine rotates backwards.

L is the reversing-link; it has the form of an arc of a circle whose radius equals $O c$. The link-block W forms the connection between the valve-stem V and link L, and makes it possible for L to be moved through a distance $f d$.

There is a joint between E and L at d, and another between F and L at f. N is a bell-crank, and is rigidly connected to the "tumbling" shaft R, which is held in position by means of bearings. It is also jointed, as shown, to the lifting-rod M and the reach-rod o; M is connected to the center c of the link L, and the reach-rod o is connected at z to the reversing-lever P, which swings about Q as a center, and when moved is caught and held in its position by the spring-latch x catching in the notches of the sector T. S is simply a counterbalance which balances the weight of the other parts about R as a center.

2087. By a movement of the reversing-lever P, through the length of the sector from the notch at x to the notch y, the point f is brought in line with a, and so changes the relative position of the slide-valve (that is, of the port-openings, etc.), that the engine can no longer rotate as indicated by the arrow, but must reverse its direction in consequence of the full steam-pressure being brought to bear on the opposite side of the piston, as a result of this movement of the valve. Another important point, in connection with this link-motion, is the fact that if the reversing-lever is moved and secured so as to bring a between d and c, the valve-travel will be reduced, and the admission-port opening diminished, directly as the distance between

a and *c*, in *a*'s new position.   When *c* reaches *a*, there will
be no travel of the valve, and for points between *c* and *f*
the valve-travel will again increase directly as the distance
between *a* and *c* increases.

This means that, as in the case of an automatic governor,
we can adjust the steam-supply to the load on the engine
for either forward or backward rotation of the crank by a
simple movement of the reversing-lever *P*, which, in this case,
operates the reversing-gears of both cylinders in Fig. 687.

2088.   This class of engines, as a rule, however, are
governed by hand, by making use of the throttle-valve
shown at *J*, and operating it by the lever *K* to check the
flow of the steam-supply, as in the case of the throttling
governor.   *H* is the main steam-supply pipe, and steam is
admitted to both cylinders at the same pressure through
the branch pipes *I* and *I'*.

2089.   Hoisting-engines are said to be **first-motion
engines** when the drum is fastened directly on the crank-
shaft, as shown in Fig. 687, and **second-motion engines**
when the rotary motion is imparted to the drum through
the medium of a small gear-wheel fastened on the crank-
shaft, which meshes with a large gear-wheel on the drum-
shaft.

HAULAGE-ENGINES.

2090.   Haulage-engines, as in the case of hoisting-
engines, usually consist of two single-cylinder engines of
exactly similar dimensions, taking steam from the same
source, and at the same pressure.   They are placed side by
side, transmit power through the same shaft, and have their
cranks at right angles to each other.   There are, however,
slight differences in the conditions under which these engines
operate in the "tail-rope" and "endless-rope" systems,
which necessitate slight differences in their construction
and operation.

2091.   The **tail-rope haulage-engine** is one of
exactly the same type as the hoisting-engine already

described in Fig. 687, except that the winding drums are usually parallel instead of conical. They should be reversible, and, since their speed varies, are usually governed by a throttle-valve operated by hand. They are also supplied with a suitable brake.

2092. An **endless-rope haulage-engine** is shown in Fig. 689. The engines A and B are connected to the
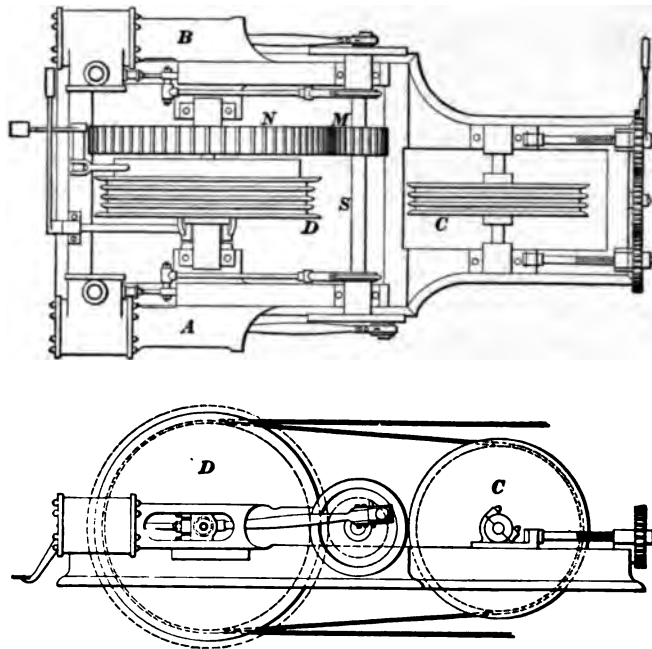
FIG. 689.

same shaft S, and transmit a rotary motion to the drum D through the gear-wheels M and N, an arrangement which constitutes *a second-motion engine*. The wire haulage-rope passes first over the drum D, and is then carried over the drum C, and, after being wound over them both, it passes off the larger drum. Since the rope is endless, that is, since its ends are spliced and it passes over a wheel at the other end of the line, there is no necessity of reversing the

engine; therefore, it is operated at a constant speed and is regulated by a throttling or automatic governor.

FAN-ENGINES.

2093. Fan-engines do not of themselves form a separate and distinct class which may be considered under this head on account of any marked peculiarity of construction which they possess. We may employ, in the driving of a fan, any engine which is capable of developing the necessary amount of power to operate the fan at the required speed. It is, therefore, evident that such an engine may be of either the simple, duplex, compound, or other form, its type being usually determined by a careful consideration of the power it is to develop and the pressure under which it is to be operated.

COMPOUND ENGINES.

2094. Compound engines are those having two cylinders of which the working lengths are the same, but the diameter of one, the **high-pressure cylinder,** is less than that of the other, the **low-pressure cylinder.** In these engines the expansion of the steam is only partially effected in the high-pressure cylinder, and on being exhausted from it, passes into an intermediate chamber which serves as a reservoir, called the *receiver*, from which the low-pressure cylinder draws its supply of steam. In the low-pressure cylinder the expansion of the steam is continued and completed; and from here it either passes into the open air or into a condenser.

2095. The chief advantage of compounding is that a greater range of expansion can be obtained than is economical in a single cylinder. With a great range of expansion there is a correspondingly great difference in the temperature of the steam from the boiler and the temperature of the exhaust. The walls of the cylinder are alternately heated by the hot steam, a part of which condenses by the process, and cooled by the exhaust. The heat taken from the cylinder walls

and carried away by the exhaust is almost a total loss.   By
allowing the steam to expand successively in two or more cyl-
inders, the range in temperature in each cylinder is reduced.
This reduces the quantity of steam condensed in the cylin-
der and the quantity of heat carried out by the exhaust.

2096.   The **tandem-compound engine,** shown in
Fig. 690, is one of the most common types of the stationary
compound engine.   In this type the high-pressure cylinder
a is usually placed directly behind the low-pressure cylin-
der *b*, both pistons being connected to the same piston-rod.
The exhaust of the high-pressure cylinder is carried in any
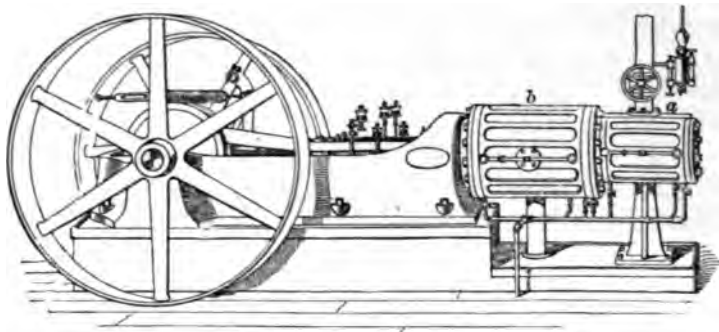
FIG. 690.

convenient manner to the low-pressure cylinder, but the
more direct the conducting passages are, the better.   This
form of construction has one great advantage of furnishing
a comparatively cheap method of compounding, as the
extra cost is only a little more than that of the additional
cylinder and its valve-gear.   Being compact, it also takes up
but very little more room than a single-cylinder engine.

2097.   Cross-compound engines are those in which
the two cylinders, each being in itself a complete engine,
just as in the case of the duplex hoisting and haulage en-
gines already described, are placed side by side and have
their cranks connected at right angles on a common shaft.
In this case, as above, the steam from the high-pressure
cylinder is exhausted into a receiver or chamber from which

the low-pressure cylinder draws its steam-supply without seriously affecting the working of the steam. This class of engine has the advantage over the tandem type of running much smoother on account of the more perfect balancing of the rotating parts. It is generally used in large constructions where the tandem type would not be practicable. In compound engines, the initial steam-pressure ranges from 60 to 125 pounds per square inch, with ratios of expansion varying from 3 to 11.

2098. Triple-expansion engines are three-cylinder compound engines. In these, high initial pressures of from 120 to 250 pounds per square inch and ratios of expansion varying from 9 to 27 are used. As in the case of the compound engine, the steam passes through each of the three cylinders of the triple-expansion engine before being finally expelled. As a general rule, engines of this type are employed only where a large amount of power is required.

2099. Single-acting engines are those which take steam during only one of the two strokes of a revolution; that is, steam is admitted to the cylinder during the forward stroke of the piston, but is shut off during the return stroke.

AIR AND AIR COMPRESSION.

PNEUMATICS.

INTRODUCTION.

2100. In order to understand the various operations of
tunneling, rock-drilling, pumping, mine ventilation, etc.,
which depend for their success upon the physical properties
of air, a knowledge of the leading principles of the proper-
ties of air and gases is necessary. That branch of me-
chanics which treats of the physical properties of air and
gases is called **Pneumatics.**

2101. The most striking feature concerning gases is
that, *no matter how small the quantity may be, they will
always fill the vessels which con-
tain them.* If a bladder or foot-
ball be partly filled with air,
and placed under a glass jar
(called a receiver), from which
the air has been exhausted, the
bladder or football will imme-
diately expand, as shown in
Fig. 691. The force which a
gas always exerts, when con-
fined, on the vessel which con-
tains it, is called **tension.** The
word tension in this case means

FIG. 691.

pressure, and is used in this sense only in reference to gases.

NOTE.—The student who is not already familiar with the elemen-
tary properties of air should read Arts. **2153** to **2168**, at the end of
this section, before proceeding further.

§ 20

PNEUMATIC MACHINES.

2102.   The Air-Pump.—The **air-pump** is an instrument for removing air from a given space.   A section of
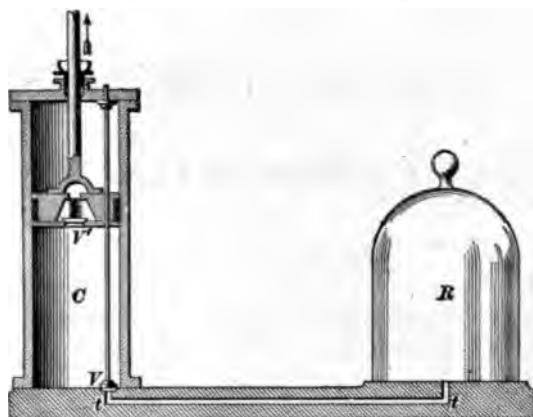
FIG. 692.

the principal parts is shown in Fig. 692, and the complete instrument in Fig. 693.   The closed vessel R is called the
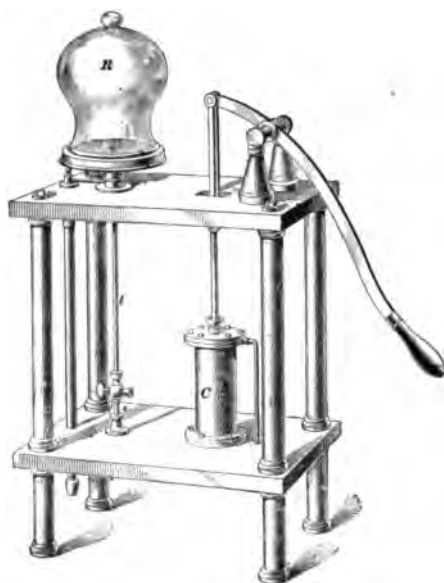
FIG. 693.

receiver, a n d t h e space which it encloses is that from which it is desired to remove the air.   It is usually made of glass, and the edges are ground so as to be p e r f e c t l y air-tight. When made in this form, it is called a **bell-jar receiver.**   The receiver rests upon a horizontal plate, in the center of which is an opening communicating with the pump-cylinder C, by means of a bent tube $t\ t$.   The pump-piston fits the cylinder

accurately, and has a valve V^7 opening upwards. At the junction of the tube with the cylinder is another valve V, also opening upwards. When the piston is raised, the valve V' closes, and, since no air can get into the cylinder from above, the piston leaves a vacuum behind it. The pressure upon V being now removed, the tension of the air in the receiver R causes V to rise; the air in the receiver then expands and occupies the space displaced by the piston, the space within the tube $t\ t$, and within the receiver R. The piston is now pushed down, the valve V closes, the valve V' opens, and the air in C escapes. The lower valve V is sometimes supported, as shown in Fig. 692, by a metal rod passing through the piston, and fitting it somewhat tightly. When the piston is raised or lowered, this rod moves with it. A button near the upper end of the rod confines its motion within very narrow limits, the piston sliding upon the rod during the greater part of the journey.

2103. Degrees and Limits of Exhaustion.—Suppose that the volume of R and t together is four times that of C, and that there are, say, 200 grains of air in R and t, and 50 grains in C when the piston is at the top of the cylinder. At the end of the first stroke, when the piston is again at the top, 50 grains of air in the cylinder C will have been removed, and the 200 grains in R and t will occupy the space R, t, and C. The ratio between the sum of the spaces R and t and the total space $R + t + C$ is $\frac{4}{5}$; hence, $200 \times \frac{4}{5} = 160$ grains $=$ the weight of air in R and t after the first stroke. After the second stroke, the weight of the air in R and t would be $(200 \times \frac{4}{5}) \times \frac{4}{5} = 200 \times (\frac{4}{5})^2 = 200 \times \frac{16}{25} = 128$ grains. At the end of the third stroke the weight would be $[200 \times (\frac{4}{5})^2] \times \frac{4}{5} = 200 \times (\frac{4}{5})^3 = 200 \times \frac{64}{125} = 102.4$ grains. At the end of n strokes the weight would be $200 \times (\frac{4}{5})^n$. It is evident that *it is impossible to remove all of the air that is contained in R and t by this method.* It requires an exceedingly good air-pump to reduce the tension of the air in R to $\frac{1}{50}$ of an inch of mercury. When the air has reached this condition of rarefaction, the valve V' will not lift, and, consequently, no more air can be exhausted.

2104. Magdeburg Hemispheres.—By means of the two hemispheres shown in Fig. 694, it can be proven that the atmosphere presses upon a body equally in all directions.   They were invented by Otto Von Guericke, of Magdeburg, and are called the **Magdeburg hemispheres.**   One of the hemispheres is provided with a stop-cock, by which it can be screwed on an air-pump.   The edges fit accurately and are well greased, so as to be air-tight.   As long as the hemispheres contain air, they can be separated without trouble; but when the air in the interior is pumped out by means of an air-pump, they can be separated only with great difficulty.   The force required to separate them will be equal to the area of the largest circle of the hemisphere in square inches, multiplied by 14.7 pounds. This force will be the same in whatever position the hemisphere may be held, thus proving that the pressure of air upon it is the same in all directions.
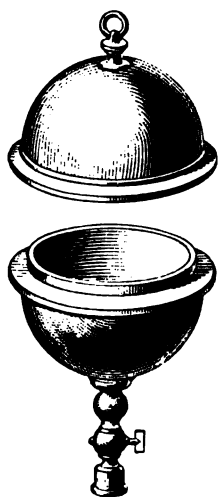
FIG. 694.

2105. The pressure of the atmosphere is very clearly shown by means of an apparatus like that illustrated in Fig. 695.   Here a cylinder fitted with a piston is held in suspension by a chain.   At the top of the cylinder is a plug *A*, which can be taken out.   This plug is removed, the piston pushed up (the force necessary being equal to the weight of the piston and rod *B*), until it touches the cylinder-head.   The plug is then screwed in, and the piston will remain at the top until a weight has been hung on the rod equal to the area of the piston, multiplied by 14.7 pounds, less the weight of the piston and rod.   If a force was applied to the rod sufficiently great to force the piston downwards, it would raise any weight less than the above to the top of the cylinder.   Suppose the weight to be removed, and the piston to be supported, say, midway of the

length of the cylinder.   Let the plug be removed and air
admitted above the piston, then screw the plug back into
its place; if the piston be shoved upwards, the farther up it
goes, the greater will be the force necessary
to push it, on account of the compression
of the air.   If the piston is of large diam-
eter, it will also require a great force to
pull it out of the cylinder, as a little con-
sideration will show.   For example, let the
diameter of the piston be 20 inches, the
length of the cylinder 36 inches, plus
the thickness of the piston, and the weight
of the piston and rod 100 pounds.   If the
piston is in the middle of the cylinder,
there will be 18 inches of space above it,
and 18 inches of space below it.   The area
of the piston is $20^2 \times .7854 = 314.16$ square
inches, and the atmospheric pressure upon
it is $314.16 \times 14.7 = 4,618$ lb., nearly.   In
order to shove the piston upwards 9 inches,
the pressure upon it must be twice as
great, or 9,236 pounds, and to this must
be added the weight of the piston and rod,
or $9,236 + 100 = 9,336$ lb.   The force neces-
sary to cause the piston to move upwards
9 inches would then be $9,336 - 4,618 =$
4,718 lb.   Now, suppose the piston to be
moved downwards until it is just on the
point of being pulled out of the cylinder.   The volume
above it will then be twice as great as before, and the
pressure one-half as great, or $4,618 \div 2 = 2,309$ lb.   The
total upward pressure will be the pressure of the atmosphere
less the weight of the piston and rod, or $4,618 - 100 =$
4,518 lb., and the force necessary to pull it downwards
to this point will be $4,518 - 2,309 = 2,209$ lb.
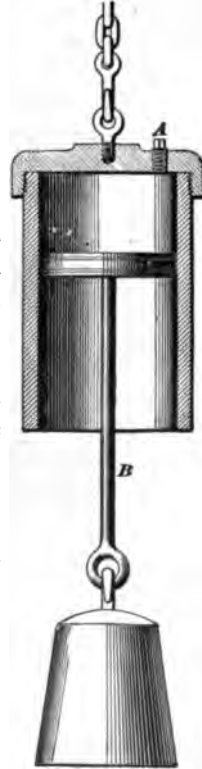
FIG. 695.

2106.   The Injector.—A section of an injector is
shown in Fig. 696.   There are many different kinds of these

instruments, but the principle is the same in all.   When
they are used for lifting water from a point below the dis-
charge orifice and forcing it into the boiler of a steam-
engine or locomotive, they depend for their lifting action
upon the creation of a partial vacuum by the action of the
steam.   In the injector, Fig. 696, *F* is the connection for
the steam-pipe from the boiler, *P* is the connection for the
pipe from the water supply, *N* is the connection to which
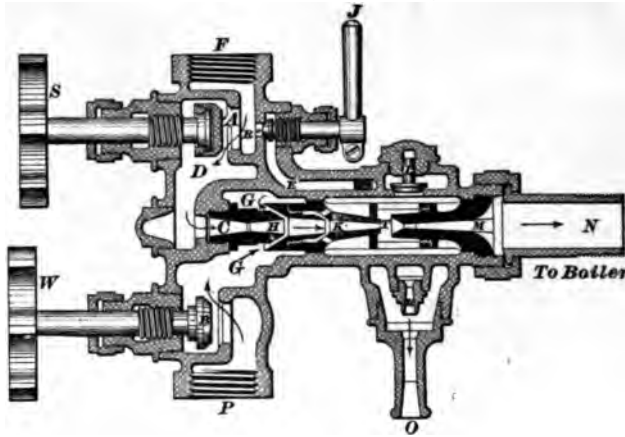
FIG. 696

the discharge-pipe leading to the boiler is attached, and the
waste water and steam are discharged through the **over-
flow** nozzle *O*.

2107. The method of operation is as follows: The
valve *B* is first opened by turning the wheel *W*; the **primer**
valve *K* is then opened by the handle *J*, thus permitting the
steam to flow through the passage *E* and a connection, not
shown in the figure, to the nozzle *u*.   From *u* the jet of
steam rushes out through *O*.   A passage connects the cham-
ber surrounding *u* with the space above valve *L*.   The jet
of steam from *u* out through *O* carries with it the air in the
chamber to which *O* is connected, thus forming a partial
vacuum in the space above *L*; the air in the passages
D, *C*, *G*, *H*, *K*, *T*, and in the water-pipe connected at *P* is
thus drawn out through the valve *L*, and a partial vacuum

is formed which permits the pressure of the atmosphere to force water through P until it finally fills the passages and flows out through L and the overflow nozzle O. As soon as water appears at O, the valve R is closed and the main steam-valve A is opened by the wheel S, thus admitting steam to the passages C, H, K. This steam draws water from G through the opening surrounding H and discharges it through K with such a high velocity that it rushes past the opening T into the nozzle M and thence into the boiler.

THE EXPANSION OF AIR AND GASES.

2108. When a gas expands, it does work; when it is compressed, work is required to be done upon the gas to compress it. Suppose that a cubic foot of air is confined in a vessel having an area of 1 square foot and a length of 5 feet plus the thickness of the piston, so that the piston can move 5 feet. Suppose the piston to be in the position shown in Fig. 697; that the absolute pressure of the volume of air enclosed in the cylinder is 100 lb. per square inch on the piston, and that the tempera-
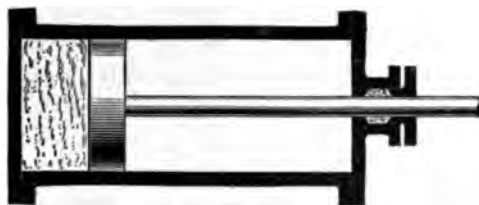
FIG. 697.

ture is 150°. Since the area of the piston is 1 square foot, the volume of the enclosed air is 1 cubic foot. Now, let this air expand, and keep the temperature constant by adding heat to it. The piston will move ahead; the atmospheric pressure upon it will be overcome through the distance it moves; the volume of the air will increase and the pressure decrease, according to Mariotte's Law. When the piston has moved 1 foot, the volume will be 2 cubic feet, and the pressure is found by the formula to be $p_1 = \dfrac{1 \times 100}{2} =$ 50 lb. per square inch. When the piston has moved 2 feet, the pressure is $\frac{100}{3} = 33\frac{1}{3}$ lb. per square inch, etc. To show, graphically, the effects of this expansion upon the pressure

and volume, two indefinite straight lines are drawn at right
angles to each other, as OY and OX, in Fig. 698. Any
line drawn from OX parallel to OY is called an **ordinate.**
Choose a convenient scale, say 1 in. = 1 cu. ft., and lay off
$OL = 1$ in. = 1 cu. ft. of cylinder volume = the volume of

FIG. 698.

air before expanding. Make $OF = 5$ in. = the total volume
after the piston has reached the end of the cylinder. Now,
choose another scale to represent the pressures, say 1 in. =
20 lb. The length of a line representing 100 lb. would be
$\frac{100}{20} = 5$ in. Lay off this distance on OY, thus locating the
point H. The pressure is 100 lb. per sq. in. throughout the
distance OL; hence, drawing HM parallel to OX, it is evi-
dent that any ordinate measured from OX to this line, with
a scale of 1 in. = 20 lb., will equal 100 lb. pressure per sq.

in. When the piston begins to move away from the position $A\,L$, the pressure begins to fall, and the volume to increase. The pressures corresponding to a number of different positions of the piston calculated by the formula $p_1 = \dfrac{p\,v}{v_1}$ are as follows:

When piston has moved ½ ft., or to d, pressure = 66⅔ lb.
When piston has moved 1 ft., or to K, pressure = 50 lb.
When piston has moved 1½ ft., or to f, pressure = 40 lb.
When piston has moved 2 ft., or to I, pressure = 33⅓ lb.
When piston has moved 2½ ft., or to h, pressure = 28¼ lb.
When piston has moved 3 ft., or to G, pressure = 25 lb.
When piston has moved 3½ ft., or to k, pressure = 22⅔ lb.
When piston has moved 4 ft., or to F, pressure = 20 lb.

At the points d, K, f, I, h, G, k, F erect ordinates, and make them equal in length to the pressure at that point, drawn to the scale of 1 in. = 20 lb.; that is, make $c\,d = 66\tfrac{2}{3}$ lb., $B\,K = 50$ lb., etc., and through the points $A, c, B, e, C,$ g, D, i, E draw the curve shown in the figure. If care has been taken in drawing this figure, any ordinate drawn from a point on the line $O\,X$ and limited by the curve will indicate exactly the pressure of the air in the cylinder when the piston is at that point. Thus, suppose it is desired to know the pressure when the piston is at the point S. Erect the ordinate $S\,R$, and measure it with the same scale that was used to measure the other ordinates; the reading on the scale will be the pressure at that point.

2109. In order to find the work done by the air while the piston was traveling from L to F, and during which the pressure fell from $A\,L$, or 100 lb. per square in., to $E\,F$, or 20 lb. per sq. in., the average pressure or mean ordinate must be known. This can be found by dividing $L\,F$, Fig. 699, into any convenient number of equal parts; in this case, 8. Erect ordinates at the points of division, thus dividing the area $A\,E\,F\,L$ into 8 parts. At the middle points of the divisions, the ordinates $1\text{-}1$, $2\text{-}2$, $3\text{-}3$, etc., are drawn and measured, the lengths being marked on the drawing.

The sum of these middle ordinates is $80 + 57.1 + 44.4 + 36.4 + 30.8 + 26.7 + 23.5 + 21 = 319.9$. Then, the mean pressure $= 319.9 \div 8 = 39.99$ lb. per sq. in. Calling the mean pressure 40 lb. per sq. in., the work which the air does in expanding from L to F at a constant temperature is equal to the area of the piston in square inches, multiplied
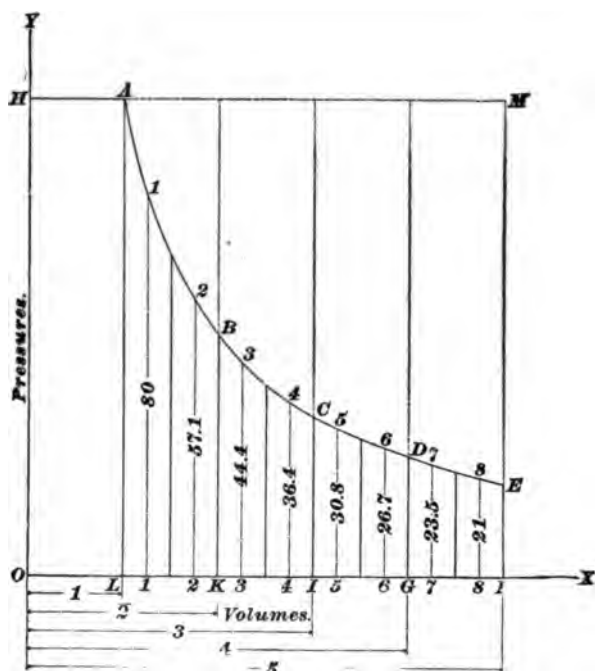
FIG. 699.

by the mean pressure per square inch, multiplied by the distance in feet through which it moves $= 144 \times 40 \times 4 = 23,040$ foot-pounds.

Suppose that the number of parts had been doubled; that is, that the line $L F$ had been divided into 16 equal parts, instead of 8, the sum of the ordinates drawn at the middle of these parts then would have been:

88.9 + 72.7 + 61.5 + 53.3 + 47.1 + 42.1 + 38.1 + 34.8 + 32 + 29.6 + 27.6 + 25.8 + 24.2 + 22.9 + 21.6 + 20.5 = 642.7.

$$642.7 \div 16 = 40.17 \text{ lb. per sq. in.}$$

$$144 \times 40.17 \times 4 = 23{,}138 \text{ foot-pounds, nearly.}$$

A sufficiently close result for all practical purposes can be obtained by dividing $A\,E\,F\,L$ into 10 parts.

2110. The curve shown in Fig. 698 is called the **isothermal expansion curve,** or the **expansion curve of constant temperature.** It is known in mathematics as the **equilateral hyperbola,** and, hence, when used on indicator-diagrams, is sometimes called the **hyperbolic curve of expansion.**

2111. If the air or gas were compressed, the action would be exactly the reverse of the expansion. Heat would have to be abstracted instead of added; the pressure would increase instead of decreasing, and the volume decrease instead of increasing.

In Fig. 700, let $E\,F$ represent the initial pressure = 20 lb. per sq. in., $O\,F$ the initial volume = 5 cu. ft. As the volume decreases, the pressure will increase, as indicated by the curve $E\,D\,C\,B\,A$, when the temperature is kept constant.

2112. Suppose that a volume of air expands from the same initial volume and pressure as in the case of Fig. 698, but that no heat is added or taken away; the temperature will fall; the pressure will fall much faster than in the case of isothermal expansion. If the air be compressed as in Fig. 700, and no heat is added or taken away, the temperature will rise; the pressure will increase much faster than in the case of isothermal compression. The formula which expresses this change of pressure and volume requires a table of logarithms in order to calculate the values; for this reason, the formula will not be given here. The work which the air can do when expanding under these conditions is considerably less than when it expands isothermally. In order to show this difference between the two cases, the pressures have been calculated which correspond to the

different positions of the piston in Fig. 698, and the fo
lowing results were obtained:

Pressure corresponding to volume O d = 56.5 lb.
Pressure corresponding to volume O K = 37.63 lb.
Pressure corresponding to volume O f = 27.47 lb.
Pressure corresponding to volume O I = 21.25 lb.
Pressure corresponding to volume O h = 17.1 lb.
Pressure corresponding to volume O G = 14.6 lb.
Pressure corresponding to volume O k = 12.0 lb.
Pressure corresponding to volume O F = 10.34 lb.
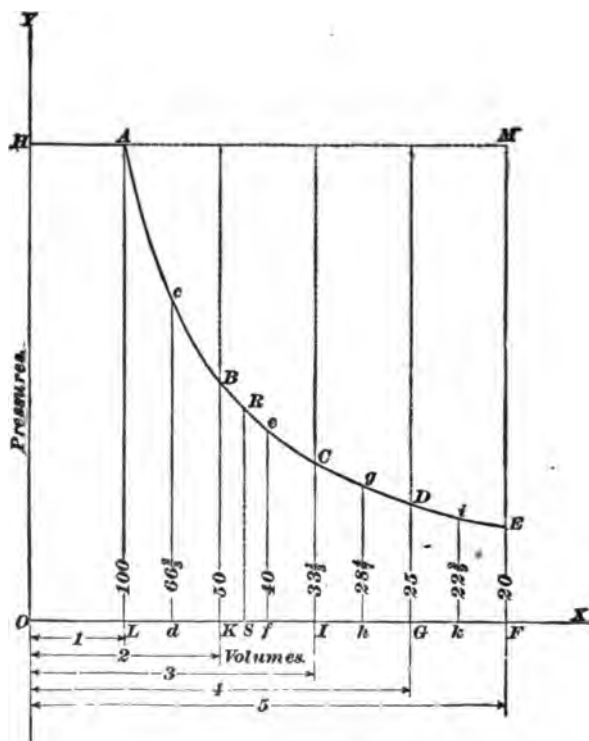
FIG. 700.

Making the different ordinates equal in length to thes
pressures and using the same scale as before, 1 in. = 20 lb
the curve shown in Fig. 701 is produced by tracing th

line *A B C D E* through these points. It will be noticed
that the area of *A E F L*, in Fig. 701, is considerably
smaller than in Fig. 698; consequently, the mean pressure is
less, and the work done in expanding is less. This was to
be expected, since, no heat being added, the temperature
must fall, and with it the pressure also. Erecting ordinates
at the middle points of these divisions and measuring them
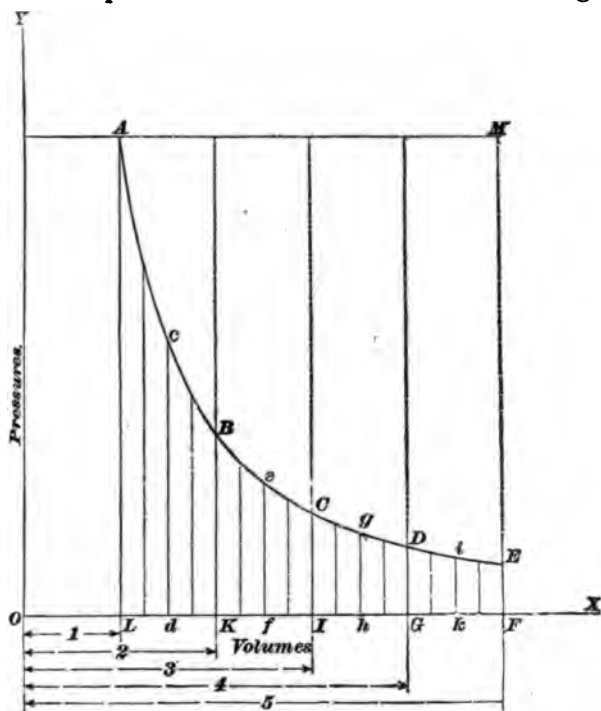
FIG. 701.

in a manner similar to the approximate method of finding
the mean pressure followed in Fig. 699, the mean pressure
is found to be:

$$\frac{73+45.5+31.9+24+19+15.5+13+11.1}{8} = 29\tfrac{1}{8} \text{ lb. per sq. in.}$$

The work done is evidently $144 \times 29\tfrac{1}{8} \times 4 = 16,776$ ft.-lb.

2113. When a gas expands without receiving or losing
any heat, the pressure falls as shown by Fig. 701, and it is

said to expand **adiabatically.** The curved line *A B C D E*
is called the **adiabatic curve.**

If the volume of air was 5 cu. ft., and the pressure was
10.34 lb. per sq. in.; that is, if the piston was at *E F*,
Fig. 701, and it was compressed to 1 cu. ft., and no heat
lost, the final pressure would be 100 lb. as before; the curve
of pressures would be the adiabatic curve *A B C D E*, as in
the case of expansion. The work which this air would do
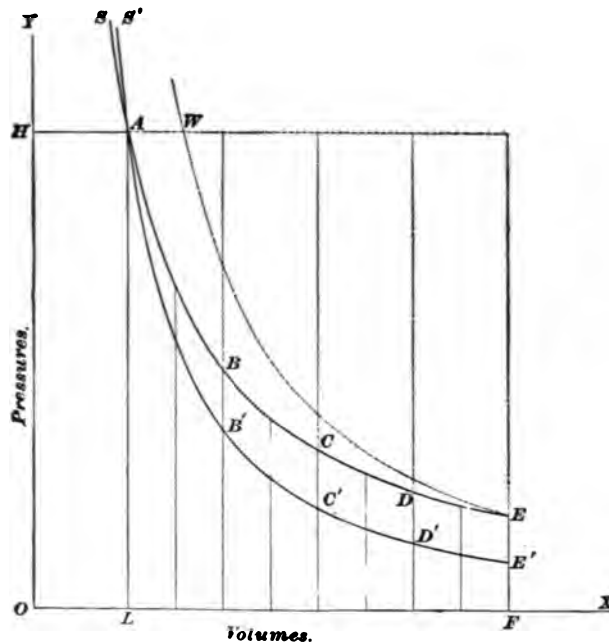
FIG. 702.

when it expanded isothermally, or at constant temperature,
was found to be 23,040 foot-pounds, and when it expanded
adiabatically, 16,776 foot-pounds, a result considerably less.
This was to be expected, since, as no heat was added, the
heat required to do the work of expansion had to be taken
from the gas, thus reducing its energy and the amount of
work that it could do. To better show the effects of iso-
thermal and adiabatic expansion, the two curves shown in
Figs. 698 and 701 are drawn together in Fig. 702. Here.

$S\,A\,B\,C\,D\,E$ is the isothermal curve of expansion or compression, and $S'\,A\,B'\,C'\,D'\,E'$ is the corresponding adiabatic curve. If 5 cu. ft. of air having a tension of 20 lb. per sq. in. be compressed isothermally, the curve of compression would follow the line $E\,D\,C\,B\,A\,S$, while, if compressed adiabatically, the initial tension and volume being the same, it would follow the dotted line $E\,W$. Hence, if the air was thus compressed to 1 cu. ft., it is easy to see that the work required would be far more for adiabatic compression than for isothermal compression.

2114. The mean pressure or ordinate of the adiabatic curve $A\,B'\,D'\,E'$ may be calculated directly, without drawing the curve and measuring the mean ordinates of the equal parts, by the aid of the following formula, which gives the area (A) of the space $A\,B'\,D'\,E'\,F\,L$:

$$A = \frac{p\,v - p_1 v_1}{.41}. \qquad \textbf{(147.)}$$

Here, p and p_1 are the greater and lesser pressures, and v and v_1 their corresponding volumes. For example, the pressure corresponding to a volume of 5 cu. ft., and denoted by the ordinate $E'\,F$, was found to be 10.34 lb. per sq. in. The greater pressure was 100 lb. per sq. in. and the corresponding volume 1 cu. ft.; hence, the area $A\,B'\,C'\,D'\,E'\,F\,L$ is

$$\frac{p\,v - p_1 v_1}{.41} = \frac{(100 \times 1) - (10.34 \times 5)}{.41} = 117.805 \text{ units.}$$

What these units are when the formula is applied to any particular figure depends upon the scale of pressures and volumes used. The mean ordinate can now be found by dividing this area by the number of cubic feet of volume represented by the length $L\,F$, in this case 4; thus, $\dfrac{117.805}{4} =$ 29.45125 lb. per sq. in. = mean ordinate. Since the area of the piston is 144 square inches, and as it moves 4 feet, the work it can do is 29.45125 × 144 × 4 = 16,964 foot-pounds. The previous calculation gave 16,776 foot-pounds, a difference of 188 foot-pounds. The difference is so slight, compared

with the whole work done, that the results are practically the same.

2115. A little thought will show that the work done is directly proportional to the areas, and that the areas themselves may be considered as representing the work done on the piston during one stroke. The mean pressure was just now found to be 29.45 lb. per sq. in. Since every inch of length on any ordinate in Fig. 702 represents a pressure of 20 lb. per sq. in., the actual length in inches of the mean ordinate is $29.45 \div 20 = 1.4725$ inches. The length of the area is 4 inches, and the actual area is $1.4725 \times 4 = 5.89$ square inches. Now, if, in any diagram of this kind, the actual area be multiplied by the scale of pressures (in this case 20 lb. per in.), then, by the scale of volumes (in this case 1 in. $= 1$ cu. ft.), and, finally, by the area of 1 square foot in inches, or by 144 square inches, the result is the work. Hence, in this case the work $= 5.89 \times 20 \times 144 = 16,963$ foot-pounds, the same result as before. The work is represented by the areas, and the ratio of any two areas is the same as the ratio of the works.

2116. A study of the curves $EDCBA$ and EW in Fig. 702 will show why the walls of air-compressors are cooled. Suppose that EF represents a pressure of 15 lb. per sq. in., instead of 20 lb. as formerly. This is about the pressure of the atmosphere, and, consequently, the initial pressure in the air-compressor cylinder. If the air was not cooled while being compressed, the pressures corresponding to the various volumes would be given by the dotted adiabatic curve EW. The work required to compress the air to a volume of 1 cu. ft. will, of course, be far greater than that required to compress it isothermally. The cooling water tends in a measure to keep the temperature constant, so that the curve of compression follows approximately the line $EDCBA$. The extra work needed when the air is compressed adiabatically is entirely used in heating the air, which subsequently cools down to the temperature of the external air. This excess work is, therefore, entirely lost, since, when

the air cools, its pressure falls. Other things being equal, the nearer the compression curve follows the isothermal curve the more efficient is the machine.

AIR-COMPRESSORS.

INTRODUCTORY.

2117. The reasons for using compressed air are many. It can be used for driving any machine ordinarily driven by steam or water without the attending loss through condensation when steam is transmitted in long pipes, and with but a slight loss through friction, compared with that sustained by water. If the pipes are of reasonably large diameter compared with the amount of air consumed, the friction loss may be disregarded. For example, 200 cubic feet of air per minute at a pressure of 60 pounds per square inch can be transmitted through 1,000 feet of 3-inch pipe, with a loss of less than 1 pound per square inch in pressure, or through a 6-inch pipe the same distance, with a loss of less than ¼ pound per square inch. Five thousand cubic feet of air per minute at 60 pounds pressure per square inch can be transmitted through a 10-inch pipe 1,000 feet, with a loss of pressure of only 1¼ pounds per square inch.

The Lehigh and Wilkes-Barre Coal Company operate the pumps in their Nottingham shaft by means of compressed air. The air is transmitted one mile through a 12-inch pipe, the gauges indicating 45 pounds at both ends.

At the Jeddo tunnel, near Hazleton, Pa., air at 60 pounds pressure was conveyed 10,860 feet through a 5¾-inch pipe. The amount transmitted was so small, compared with the size of the pipe, that the gauges placed at both ends of the pipe indicated 60 pounds. Consequently, when deciding upon the merits of a compressed-air plant, the loss through friction may be neglected, it being only a question of the size of the pipe.

2118. The first cost of the apparatus is low compared with other methods of operating mining-tools. The air

exhausting from the drills, pumps, etc., assists in venti-
lating the mine. The apparatus is durable, of light weight,
and occupies but little space. The air outside of the com-
pressor from which its supply is drawn is termed **free air.**
The lower the temperature of the free air, the greater will
be the efficiency of the compressor. For example, suppose
the temperature of the free air to be 0° F., and that a cubic
foot is compressed adiabatically to 3 atmospheres gauge; its
temperature will be about 225°, an increase of 225° in the
temperature. If the free air at a temperature of 60° be com-
pressed adiabatically to 3 atmospheres gauge, its temperature
will be about 315°, an increase of 315° − 60° = 255°; if com-
pressed to 3 atmospheres gauge from a temperature of 100°,
its temperature after compression will be about 380°, an in-
crease of 380° − 100° = 280°. If the air be compressed to
7 atmospheres in a manner similar to that just described, and
from corresponding temperatures of the free air, the tem-
peratures after compression will be about 380°, 490°, and
560°, respectively. The corresponding increases in tem-
perature will be 380°, 490° − 60° = 430, and 560° − 100° =
460°. If the compression be carried farther, there will be
still greater differences. It is thus plainly seen that the
lower the temperature of the air when it enters the cylin-
der, the more efficient will be the machine.

2119. For mining purposes, few compressors condense
(compress) the air to less than 3 atmospheres (44.1 pounds
per square inch) gauge pressure, or exceed 7 atmospheres
(102.9 pounds per square inch) gauge pressure. As was
shown in the preceding pages, air becomes heated when
compressed, and the pressure increases very rapidly. If no
means are provided for cooling the air while being com-
pressed, so that it will be kept at, as nearly as possible, the
same temperature it had on entering the cylinder, the curve
of compression will follow the adiabatic curve. When the
compressed air is used near the point where the compressor
is situated, it occasions no particular loss; but when the
compressor is situated a thousand feet or more from the

point where the air is used, and the air has to be transmitted to that point through 1,000 feet or more of pipe, it cools down to the temperature of the outside air; its pressure falls in consequence of this loss of heat, and there is a very considerable loss of power. Add to this the friction of the engine and compressor, a slight loss through friction of the air in the pipe, and the loss through leakage; the result is that, even when the air has been cooled to a greater or less extent, according to the type of compressor, the efficiency averages about 50%, being above that in some plants and below in others. By *efficiency* is meant the ratio of the work obtained from the air to the work done in compressing it. The first can be obtained when the pressure and amount of air used in a given time is known, and the last is found from the indicator-card of the steam-engine, or by other means if some other motor is used. Thus, suppose that the indicated horsepower of the steam-cylinder is 23.45 and the power obtained from the compressed air is 13.8 horsepower,

the efficiency would be $\dfrac{13.8}{23.45} = .5885$, or 58.85%.

Unless the air is cooled during compression, or some other device (to be described farther on) is employed, the efficiency will fall below the 50% average given above.

TYPES OF AIR-COMPRESSORS.

WET COMPRESSORS.

2120. There are two systems in use by which it is attempted to absorb the heat developed during compression. They are so different in their methods of cooling and in the results obtained that it is usual to make two distinct classes of them, viz., *wet compressors* and *dry compressors*.

2121. A **wet compressor** is one in which the water is introduced directly into the air-cylinder, and thus brought into contact with the air. It is made in two forms; in one, the water is injected into the cylinder in the form of a finely divided spray, thus mixing thoroughly with the air; in the

other form, the cylinder is filled with water on both sides of the piston, the air being admitted above the water, and is compressed by the water rather than by the piston.

2122. In Fig. 703 is shown a **Dubois & Francois high-speed double-acting water-injection air-compressor.** In order to more clearly show the valve arrangements, etc., a section of the air-cylinder only is given. The piston rod R is connected at its other end to the piston of a
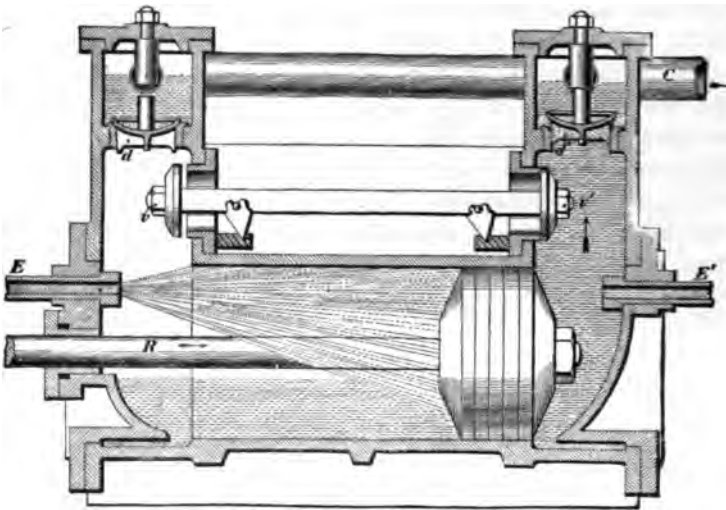
FIG. 703.

steam-engine, which affords the power needed for the compression. Let the piston be moving in the direction of the arrow. The valve c is open, and the air is following the piston during the stroke. The water is being injected through the two inlets E and E'. On the right-hand side of the piston, the water is being shoved upwards to fill the clearance space between the piston and the discharge-valve d'; the air which occupied this space is compressed, and keeps the inlet-valve c' against its seat. When the air reaches a certain pressure, which can be fixed to suit the purpose for which the air is to be used, the discharge-valve

d' is raised, and the air passes out and is discharged through the delivery-pipe *C* into a conduit. Any excess of water is also discharged through the valve *d'* into the conduit, but is collected and forced back into the cylinder through the nozzles *E* and *E'*.

Suppose the piston to be on the return stroke. The valve *d'* falls; the weight of the water causes it to fall and follow up the piston, leaving a vacuum behind it. The pressure of the atmosphere against the left side of the valve *v'* forces it to the right, and, with it, the valve *v* against its seat. The air then flows in and follows up the piston on the right side and is compressed on the left side, the operation being repeated exactly as before described. It will be noticed that both discharge-valves open into the same delivery-pipe *C*. This is called a **double-acting compressor,** because the air is compressed on both sides of the piston, that is, twice during each revolution of the crank-pin.

2123. In Fig. 704 is shown an elevation and section through the air-cylinder of a **Burleigh single-acting vertical air-compressor.** Only one cylinder is seen in the cut, but there are two more behind the one shown—an air-cylinder, and a steam-cylinder to drive the compressor. The cranks of the air-cylinders are set directly opposite each other, so that they are on the opposite dead-points at the same instant. The air is compressed during the upward stroke, and admitted during the downward stroke. The piston has a large valve *V* in it which is raised during the downward stroke, allowing the free air to enter and fill the cylinder; at the same time, water is injected through the pipe *S*. The water is not sprayed in during compression, as in the compressor previously described. It nearly fills the clearance space, and cools the air somewhat by reason of cooling the cylinder walls and compelling the air to come into contact with it when resting on top of the piston during the up stroke. The discharge-valve *d* is raised when the pressure reaches the desired point, the air passing out through the delivery-pipe *C*.
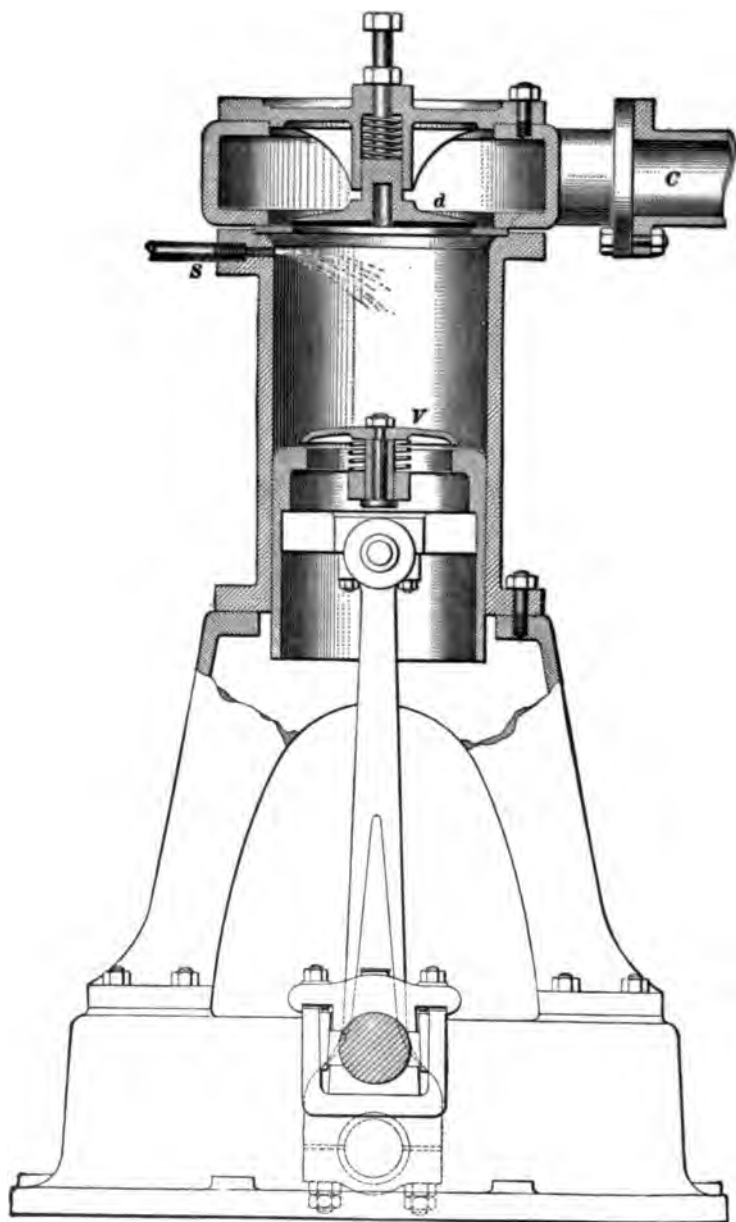
FIG. 704.

2124. In direct-acting air-compressors where the air-cylinder is behind the steam-cylinder, and both air-piston and steam-piston are connected to the same rod, the point of greatest compression is the point where the steam pressure is nearing its lowest point, unless there is no cut-off, and the steam follows the piston for the full stroke at initial pressure. This, however, is very wasteful of steam, and with a single steam and air cylinder in line, the compressor requires a heavy fly-wheel in order to have a cut-off. In the Burleigh compressor, the crank of the steam-cylinder is so set that when the cranks of the air-cylinder are on their dead-points, the steam-piston will have advanced only about $\frac{1}{8}$ of its stroke. The steam-cylinder is, of course, double-acting, and by this arrangement the full steam pressure acts upon the air-piston at the point of greatest compression.

2125. Another type of the wet compressor is shown in Fig. 705. The water surrounds the piston on both sides and partly fills the chambers above. Suppose the piston to
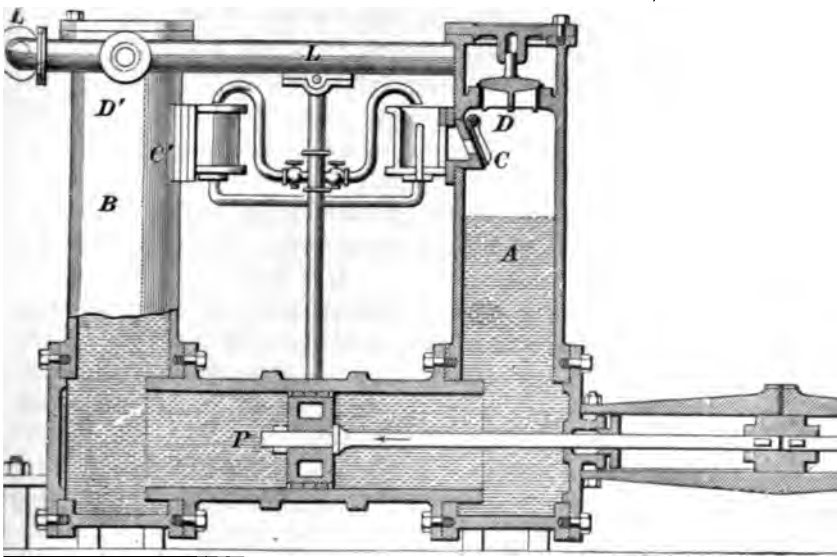
FIG. 705.

be moving to the left; the water in the chamber A falls,
following the piston, and leaves a vacuum behind it. The
pressure of the free air causes the valve C to open, and the
chamber A is thus filled with air at atmospheric pressure.
In the mean time, the air in B is compressed, and when the
desired pressure is reached, the valve at D' (similar to D) is
raised, and the air is discharged into the delivery-pipe L.
When the piston begins its return stroke, the pressure of
the air in the delivery-pipe causes the valve D' to close; the
inlet-valve at C' opens, and the foregoing process is repeat-
ed. The chambers A and B, the valves C and C', and also
the valves D and D', are exactly alike.

2126. The type of air-compressor last described is
much inferior to the water-injection type. It has all of the
disadvantages of the wet-compressor class, and, in addition,
it will not deliver as cool air as a compressor in which the
water is injected in the form of a fine spray and thoroughly
mixed with the air. The following are the principal objec-
tions to the wet-compressor class:

I. The impurities in the water. The water may be strong-
ly acid or strongly alkaline, and act chemically upon the
metal surfaces, thus gradually destroying them. It may
contain dirt and grit, and thus wear out the cylinders, pis-
tons, packing, etc., very rapidly.

II. The water renders lubrication difficult, owing to the
fact that oil floats on its surface. This also increases the
wear of the parts.

III. The water absorbs a considerable portion of the com-
pressed air, which is, of course, entirely lost.

IV. There is a loss of power, owing to the inertia of the
water, the engine being required to put it in motion and
bring it to rest during every stroke.

V. The speed of the compressor is limited to that of a
water-pump, the average piston speed of which is about 100
feet per minute.

VI. The greatest objection is that, when the wet air is
used expansively, the moisture freezes in the cylinder and

exhaust-pipes, owing to the temperature of the expanding air falling many degrees below zero. The ice collects, and in many cases stops the engine. This last objection has been almost eliminated, owing to the means employed to rid the air of its moisture when in the receiver.

2127. The arguments in favor of the wet compressor, particularly of the injection type, are that the air is compressed nearly isothermally, and there is no loss, owing to the large clearance space between the piston and the valves when the piston is at the end of its stroke. Diagrams taken from the cylinders of a wet compressor of the injection type gave the following results: The work expended in compressing 10.76 cubic feet of air to 4.21 atmospheres was 38,128 foot-pounds. Compressed isothermally, the work would have been 37,534 foot-pounds, the difference being 594 foot-pounds, or a loss of 1.6%. Compressed adiabatically, the work would have been 48,158 foot-pounds. The temperature of the air on entering the cylinder was 50° F., and, on leaving, 62° F., an increase of only 12° F. Had the air not been cooled, the resulting temperature would have been 352° F., and the increase, 352° — 50° = 302° F.

DRY COMPRESSORS.

2128. In Fig. 706 is shown a **Clayton double-acting water-jacketed air-compressor.** The steam and air cylinders are in line with each other, but are situated on opposite ends of the bed-plate, with the crank-shaft and fly-wheel between them. The peculiar construction of the cross-head permits this. The two parts C, C of the cross-head are joined by two strong rods D, of which only the upper one is visible. The bottom of the bed is planed, and the cross-head slides upon it as a guide. H is the air-delivery pipe. The object of placing the fly-wheel in the center, and having the two cylinders opposite each other, instead of tandem, or one behind the other, is to economize room.

A section of the air-cylinder is shown in Fig. 707. Suppose the piston to be moving in the direction indicated by the arrow. The suction-valves (inlet-valves) D', which open inwards, are forced to their seats by the pressure of the air in front of them, and the discharge-valves F' are forced open for the same reason, when the air reaches the desired pressure, which is determined by the tension of the springs at E'. The air follows the arrows and discharges through H, being prevented from entering the other end of the cylinder by reason of the spring E pressing the valves F to
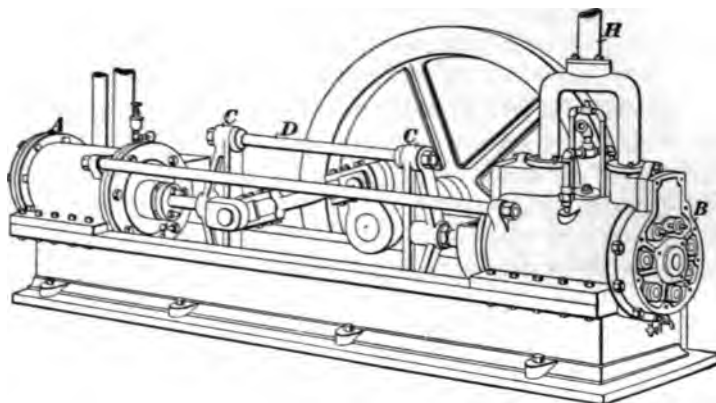
FIG. 706.

their seats. While the piston has been moving as described and leaving a vacuum behind it, the atmospheric air has forced open the valves D against the resistance of the springs C, allowing the free air to enter the cylinder. At the instant that the piston reaches the end of its stroke, the springs C draw the valves D to their seats and prevent the air retained in the cylinder from passing out. The process is again repeated, the discharge being through the valves F in the left-hand end of the cylinder.

This illustration also shows the manner of cooling the air; the walls of the cylinder are hollow, and the water enters through the U-shaped pipe K K (see also Fig. 706), flowing around the cylinder until it finally passes out through the water-delivery pipe L. The cold water cools the cylinder

walls, which, in turn, cool the air. This method of cooling
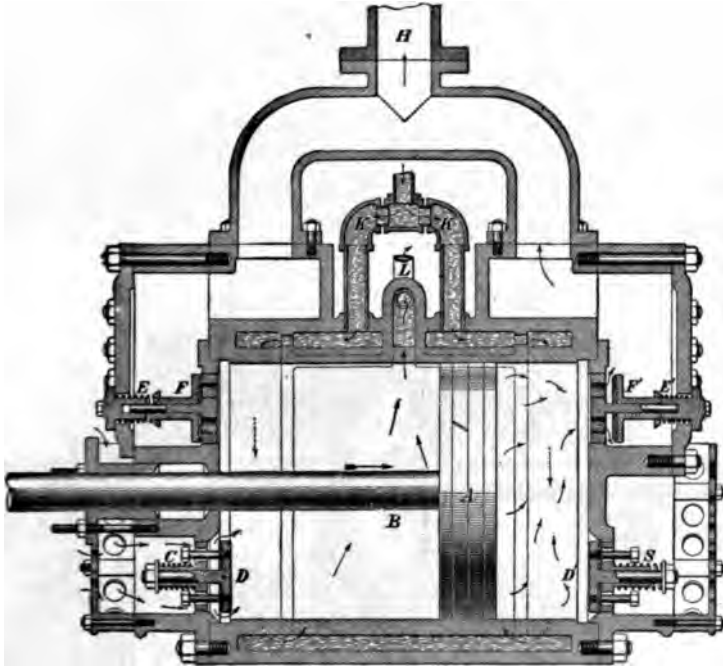
FIG. 707.

by having the water circulate around the hollow cylinder walls is termed a **water-jacket.**

2129. The clearance space has been mentioned several times in the preceding pages, as if it exerted a prejudicial effect upon the working of the compressor. To show the effect which it really produces, assume that the air has been compressed isothermally. In this case, there would be a certain loss of power, owing to heat having been produced and then absorbed by the cooling methods employed. As part of the air was not discharged, the work required to heat this air has been lost. Had the air been compressed adiabatically, the extra heat due to compression retained by the air in the clearance space is given up during the return stroke, and assists in the work of compression. The best

air-compressors give results midway between the isothermal and adiabatic compression, and when the clearance space is not excessive, the loss due to it is so small that it may be practically neglected.

2130. A section of the air-cylinder of a compressor in which the clearance is reduced to a very small amount is shown in Fig. 708. This is an **Ingersoll-Sergeant piston-inlet compressor.** The piston is cast hollow, the rod
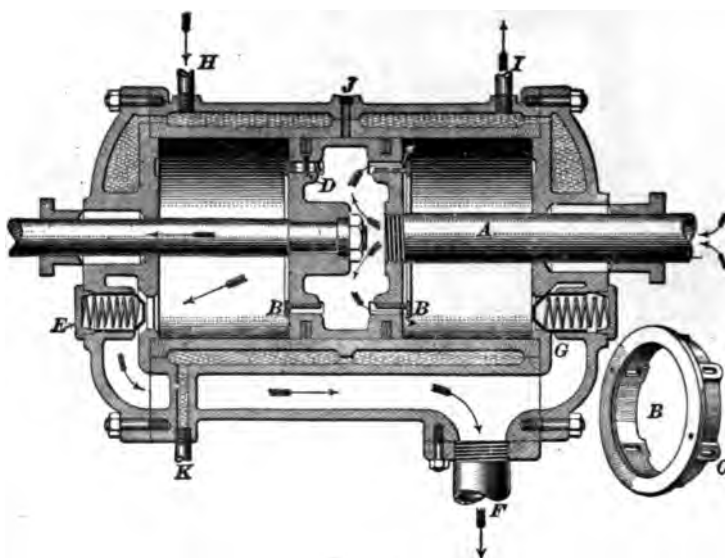
FIG. 708.

being fastened to it by means of a nut, as shown. A tube *A*, of such length that, when the piston is at the left-hand end of its stroke, its end will project through the stuffing-box on the right-hand side, is screwed into the right-hand side of the piston. The atmospheric air communicates with the hollow space in the piston by means of the tube *A*, as shown in the figure. On both sides of the piston is a ring *B* (shown in perspective at the right-hand side of the section). It will be noticed that the ring has an extension in the form of a hollow cylinder on which are lugs *C* having oblong holes. On the piston itself are taper pins *D*, which fit loosely in the

>f the lugs, so that a slight forward or backward
ient of the ring can be obtained. These rings form
et-valves, and operate as follows: Suppose that the
is moving in the direction indicated by the arrow on
;ton-rod. The right-hand side valve B is open; the
nd side valve B is closed by reason of the pressure of
npressed air acting against it. The free air enters the
through the tube A, and flows out through the right-
ide inlet-valve B into the cylinder. When the proper
re has been reached, the delivery-valve E opens, and
is discharged, following the direction indicated by the
;, out through the pipe F. When the piston reaches
d of its stroke, and reverses, the valves B tend to
ue in the former direction, according to Newton's first
motion; hence, their inertia causes the right-hand
:o be thrown against its seat, and the left-hand valve
a. The operation above described is again repeated,
that the free air now flows through the left-hand
alve, and the compressed air is discharged through G.
air is cooled by means of a water-jacket, the walls
hollow and the water flowing around them, entering
h H and flowing out through I. K is a drain-pipe
:d with a valve, and is for the purpose of draining the
from the cylinder. As there are no suction-valves in
ds of the cylinder, the greater part of the cylinder-
is also water-jacketed. Oil drops through the small
J, and lubricates the cylinder. The clearance is re-
to a very small fraction of the cylinder volume, and
npressor can be run at as high a speed as desired.

DUPLEX COMPRESSORS.

$1. With the exception of the Burleigh, all of the
essors previously described have been what are termed
;ht-line compressors; that is, the center lines of
:am and air cylinders have formed one straight line.
two straight-line air-compressors are placed side by
aving a common crank-shaft, they are called **duplex**

compressors. The cranks of this type are set at right angles, and the distribution of the power may be understood from Figs. 709 and 710. In Fig. 709, *A* and *B* are the
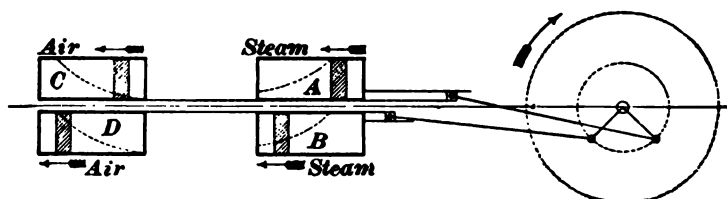
FIG. 709.

steam-cylinders, and *C* and *D* are the air-cylinders. The larger dotted circle represents the fly-wheel, and the smaller one the path of the crank-pin. The cylinder *A* has the full steam pressure on its piston; as but little power is needed in *C* at this point, the greater part of the work is transmitted through the shaft to the piston in *B*, and from thence to the air-piston *D*, where the compression has now reached its highest point. In the cylinder *B*, the steam has expanded until it is very near its terminal pressure, as indicated by the dotted expansion line. In *C*, the compression is just beginning.

In Fig. 710, the two cranks of the last figure are imagined to have turned through a quarter of a revolution. The con-
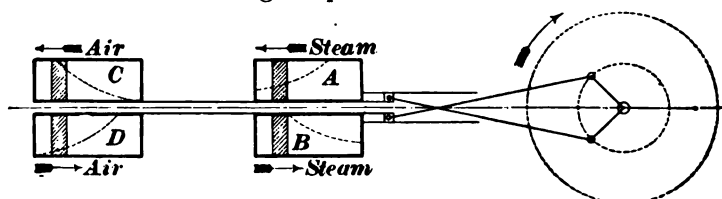
FIG. 710.

ditions are now reversed. The stroke is being completed in *A*, and just beginning in *B*; the air has reached the point of highest compression in *C*, and the compression is just beginning in *D*. The greater part of the work in *B* is now transferred to *C*. It will be seen from this, that the steam acts principally to drive the air-piston of the cylinder diagonally opposite to it.

2132. An illustration of a light **duplex compressor** made by the Rand Drill Co. is shown in Fig. 711. It is so made that it can easily be taken apart and transported on
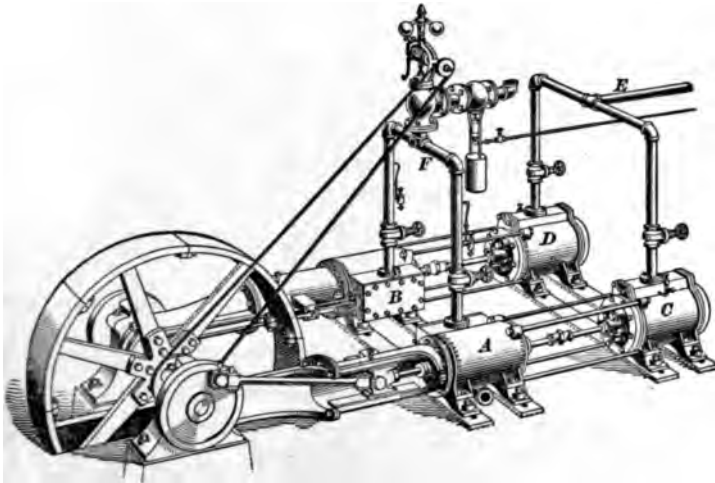
FIG. 711.

muleback. *A* and *B* are the steam-cylinders, and *C* and *D* are the air-cylinders. *E* is the air-delivery pipe, and *F* is the steam-pipe.

2133. Some of the advantages of the duplex type are the following: Since the cranks are set at right angles, the engine can not get on its dead-center. One cylinder can be detached from the other when only half the capacity of the machine is required. The power and resistance being equalized through opposite cylinders, large fly-wheels are not necessary.

2134. Some of the disadvantages are: The strains are indirect, angular, and intermittent. It is necessary, therefore, to greatly increase the strength of parts, to add a crank of increased diameter with larger bearings, and to build very much stronger foundations, since excessive strains will be brought upon the bearings should the foundations settle at any point, resulting in friction and liability to

breakage. The friction loss in the duplex type is seldom less than 15% of the indicated horsepower of the engine, while in the straight-line type it is sometimes as low as 5%.

COMPOUND COMPRESSORS.

2135. When very high pressures are desired, compound air-compressors should be used. In these machines the air is compressed usually to about 30 pounds in the first cylinder, and then delivered into the other cylinder, where it is compressed to any desired extent.

2136. A compound air-compressor built by the Norwalk Iron Works is shown in Fig. 712. *A* is the low-pressure cylinder, into which the free air is taken. *B* is the
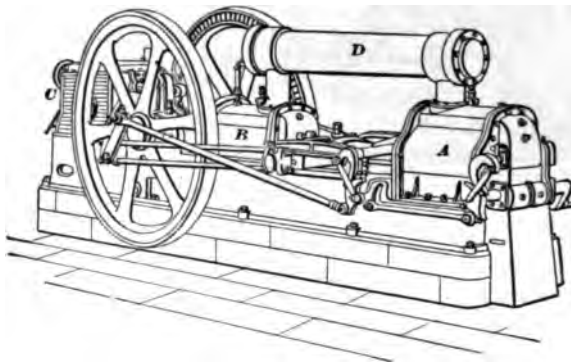
FIG. 712.

high-pressure cylinder, and *C* is the steam-cylinder, all three being in line. The two air-cylinders are water-jacketed. The air is first compressed to 30 pounds per square inch in *A*, and is then discharged through the large pipe *D*, called the inter-cooler, into *B*, where it is compressed to the required pressure. The pipe, or inter-cooler, *D* is also water-jacketed, so as to cool the air and get it into the high-pressure cylinder at as low a temperature as possible. The valves are operated by means of cams and levers, an arrangement in many respects superior to the poppet or spring valves of the previously described compressors. The inter-cooler effects quite a saving in power, this saving being
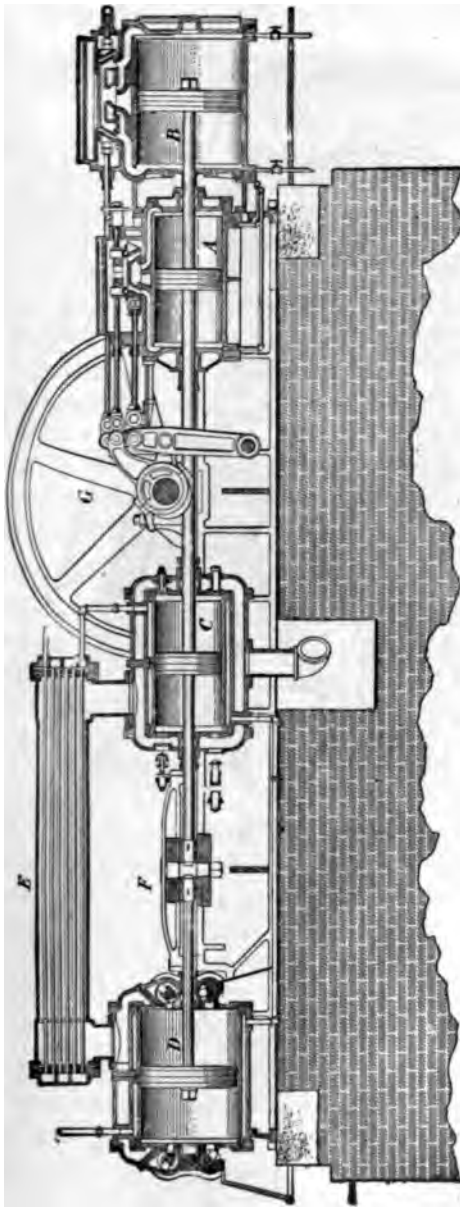
dependent upon the temperature of the air when entering the high-pressure cylinder, and also upon the ratio of the volume of the inter-cooler and clearance space outside of the discharge-valve to that of the low-pressure cylinder. The principal advantage of the compound air-compressor lies in the reduction of the clearance-space and in the equalizing of the stresses on the engine. By means of the compound air-cylinders, a pressure of 5,000 pounds per square inch can be obtained. To obtain such a high pressure as this in a single cylinder would require a very long stroke, and the loss due to clearance would be excessive.

2137. In Fig. 713 is shown a section of a compound air-compressor

FIG. 713.

24—16
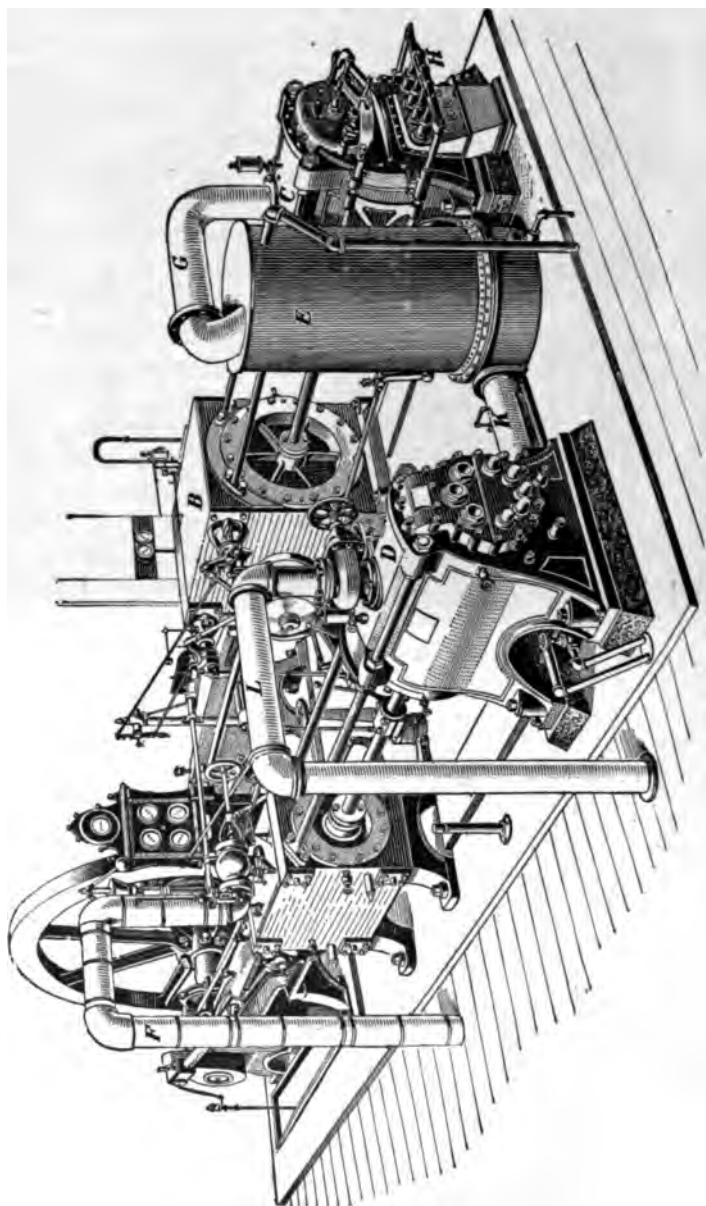
FIG. 714.

t by the same company, which differs from the other,
n the fact that it is driven by a tandem-compound
.m-engine. All four cylinders are in the same straight
. A is the high and B the low pressure steam-cylin-
; C and D are the corresponding air-cylinders. E is
inter-cooler, which is partly filled with pipes through
ch the cold cooling water circulates. These pipes
de the air-current into small streams, and enable nearly
ry particle to come into contact with the cold sur-
:, thus reducing the temperature very rapidly. F is
cross-head, and G one of the two fly-wheels. In this
ipressor, all of the advantages are obtained that are
racteristic of the tandem-compound type of engine,
:ther with those to be derived from the compound com-
ssor previously described; in other words, the consump-
i of steam is reduced, and the strains are far more
ally distributed throughout the stroke.

138. A Rand duplex-compound air-compressor
ren by a **Corliss cross-compound condensing**
am-engine is shown in Fig. 714. A and B are the
h and low pressure steam-cylinders, F being the steam-
e; C is the low-pressure air-cylinder. Air enters each
l of the cylinder alternately. The inlet-valves H are
uated positively by a combination of levers and yokes,
springs being used. The air is here compressed to
iost 30 pounds, and then discharged through the pipe G
o the inter-cooler E, where the temperature is reduced
means of coiled pipes through which cold water circu-
es. From the inter-cooler, the air is conducted through
: pipe K into the high-pressure cylinder D, where it is
'ther compressed to the required pressure and discharged
'ough the pipe L into the receiver.

2139. Fig. 715 shows a plan and side elevation of a
plex air-compressor driven by water. The fly-wheel F
s a large number of cup-shaped projections on its rim.
ie water is conducted to the wheel by the pipe A, dis-
irging at a high velocity and striking the cup-shaped

vanes, thus causing the wheel *F* to turn. The wheel imparts its motion to the cross-head and piston by means of the connecting-rod *B*, thus compressing the air. These machines have a high efficiency when properly designed, and are very simple in their construction. Where a natural head of water, say of 50 feet or more, is available, they can be used to a great advantage.
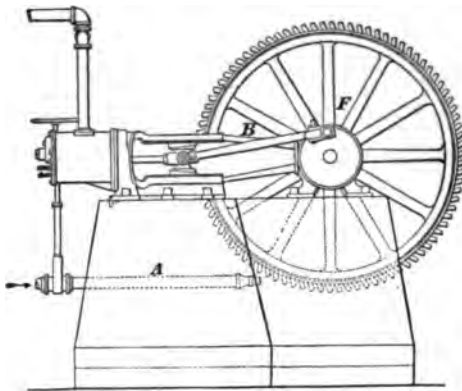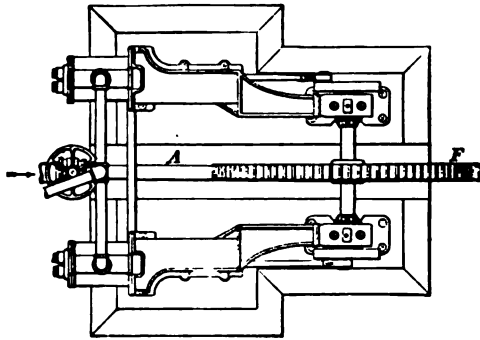
FIG. 715.

RECEIVERS.

2140. In all cases governing the use of compressed air, there should be a **receiver** placed within at least fifty feet of the compressor.   The receiver acts not only as a reservoir, but also corrects the irregularity of air admission from
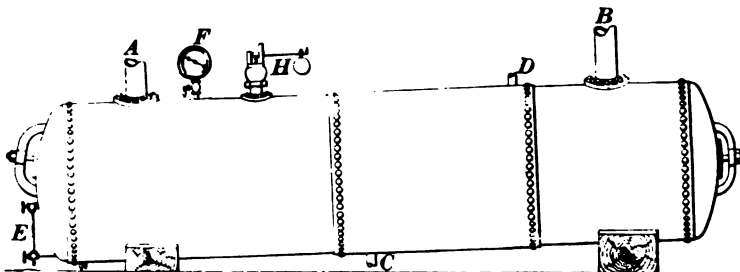
FIG. 716.

the compression-cylinder, and delivers it to the hoist, pump, or drill with great regularity of pressure, in much the same manner that a boiler delivers steam to an engine.   In design-ing receivers, when the compressed air is to be used for driving rock-drills, it is customary to allow about ten cubic feet of receiver-volume for each drill; i. e., for five drills, the volume of the receiver would be about 50 cubic feet. In all cases, the larger the receiver, the better.

2141.   In Fig. 716 is shown a **horizontal air-receiver,** and in Fig. 717 a **vertical air-receiver.**   The air enters
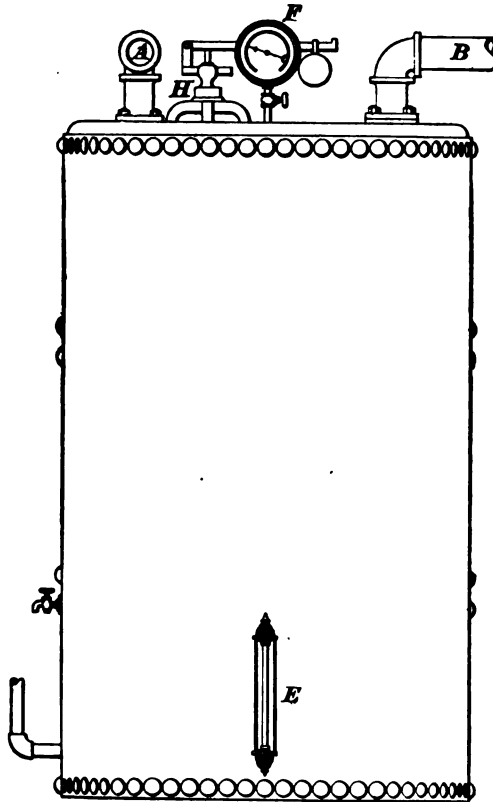
FIG. 717.

the receiver at *A*, flows through a series of pipe-coils, and

discharges through *B*. These pipe-coils are constantly surrounded by water flowing through them; this cools the air and dries it at the same time, the moisture dropping to the bottom of the coils. The glass gauge *E* indicates the amount of moisture deposited, and when it gets higher than desired, it is drained off. The cooling water enters at *C* and is discharged at *D*. *F* is a gauge to show the pressure of the air, and *H* is a safety-valve to prevent the pressure from becoming too high and making the receiver dangerous through liability to blow up.

PRESSURE-REGULATORS.

2142. In addition to a receiver, an air-compressor should be provided with a **pressure-regulator.** The pressure-regulator is to an air-compressor what a governor is to a steam-engine. Fig. 718 shows a regulator manufactured by the makers of the Norwalk compressors. It is placed on the steam-supply pipe of the steam-cylinder, and its function is to reduce the amount of steam admitted to the engine cylinder when the air pressure reaches a fixed limit in the receiver, and to admit more steam when the consumption of the compressed air has increased to such an extent that the pressure in the receiver is lowered. The less steam admitted, the slower the engine speed, and, consequently, the less the amount of air compressed; the more steam admitted, the greater

FIG. 718.

the engine speed and the greater the amount of air compressed. Hence, the amount of air compressed per minute is varied by varying the speed of the compressor.

In the figure, *A* is the body of a balanced globe-valve; that is, the pressure is the same on all sides of the valve, and a very slight force only is required to move it up or

down. Above the valve is a small cylinder B, having a pis-
ton connected to the valve below by the stem C. At the
side of this cylinder is a small spring safety-valve D, the
under side of which connects with the receiver by a pipe.
The hand-wheel E varies the tension of the spring so that
the pressure in the receiver can be adjusted as desired.
When the pressure in the receiver has reached the desired
limit, the safety-valve D allows the air to escape and pass
into the small cylinder B, beneath the piston; if no escape
was provided, the piston would be driven to the top of the
cylinder, the valve in A would be held to its seat, and the
engine stopped. To prevent this, a very fine slot is cut
in the side of the small cylinder B. When the piston rises,
it uncovers this slot, and thus furnishes an escape for the air
which is passing the safety-valve. If only a little air passes
the valve, a small part of the slot will accommodate it, and
the piston will take a low position, the speed of the engine
being then but slightly reduced. If more air escapes, the
piston will rise higher, in order to allow more of the slot to
be uncovered, and thus provide a larger opening for the exit
of the air, the engine speed being still further reduced.
That the engine may be prevented from entirely stopping,
a screw-stop F is placed on the top of the cylinder B; this
prevents the valve in A from closing more than is sufficient
to run the engine at the slowest speed that will carry it over
the dead-centers.

INDICATOR-CARDS.

2143. In Fig. 719 are shown two indicator-cards, one
taken from the steam-cylinder and the other from the air-
cylinder. The most striking characteristic of these cards is
that the pressure of the discharged air is considerably high-
er than the initial pressure of the steam. The area of the
two cards is very nearly the same, that of the air card being
a little less; this shows that the work done in both cylinders
is practically the same, the extra work shown by the steam
card being used to overcome the engine friction. The ex-
tra energy of the steam in the first half of the stroke where

it overcomes but a slight resistance is stored in the fly-
wheel and reciprocating parts, and given out when needed at
the end of the stroke. The reason that the upper line of
the air card is wavy instead of being straight is that the
discharge-valve has a constant tendency to close, and is held
open only by the pressure of the air in front of it. This
varies somewhat towards the end of the stroke, and causes
the valve to have a slight to-and-fro movement, which pro-
duces the wavy line. If the valve were worked in such a
manner that it could open at only one point and be closed
at another, like the valves of an engine-cylinder, the line
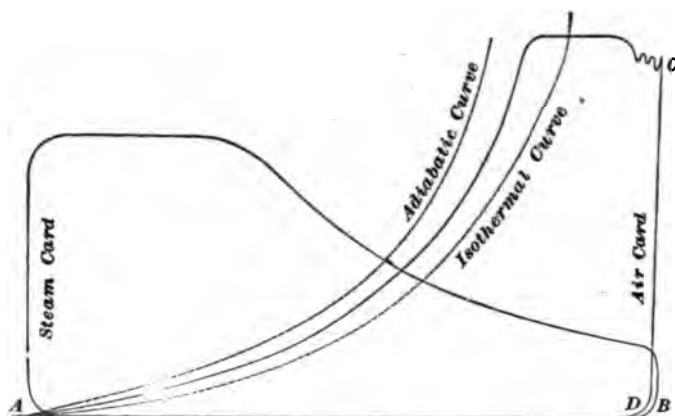
FIG. 719.

would be straight. The card also shows that the line CD
is very nearly perpendicular to the atmospheric line AB,
and also that it is practically straight. This indicates that
the clearance is very small, since, if there were air of a high
pressure behind the piston when it began the return stroke,
it would expand and cause the line CD to curve, correspond-
ing to the compression-line of a steam-engine cylinder dia-
gram. This effect is illustrated in Fig. 720. The two
dotted lines in Fig. 719 are the adiabatic and isothermal
compression-curves, the actual compression-line falling
about half way between them, owing to the water-jacket.
These air diagrams serve not only to show the work actually
needed in the air-cylinder, but also to determine the volume

of the free air actually compressed. Theoretically, this volume is equal to the area of the piston, multiplied by the length of the stroke. Actually, however, owing to imperfections in workmanship, the air does not begin to compress at the instant the piston begins its stroke. The point where the compression begins is indicated on the diagram by the point A, where the compression-curve begins to leave the atmospheric line $B\,C$. The length of the stroke is proportional to the length of the atmospheric line $B\cdot C$. The ratio
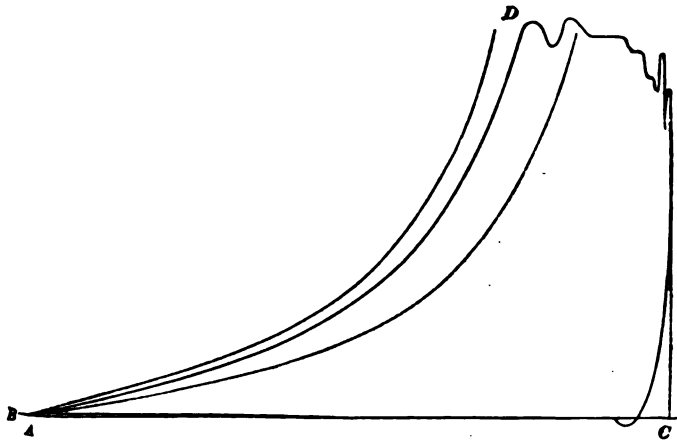
Fig. 720.

of the lengths of $B\,A$ to $B\,C$ will give the percentage of theoretical volume lost. In this case, $\dfrac{B\,A}{B\,C} = .034$, nearly, $= 3.4\%$.

REHEATING COMPRESSED AIR.

2144. Air can, of course, be expanded exactly like steam when used as a motive power, and the gain through expansion is nearly as great as in the case of steam. The chief difficulty lies in the intense cold produced by air, at a high pressure and normal temperature, expanding down to the pressure of the atmosphere. Thus, if a cubic foot of air at a temperature of 60° and a pressure of 88.2 pounds be expanded adiabatically, performing work, to the pressure of the atmosphere, the resulting temperature will be 151° below

FIG. 711.
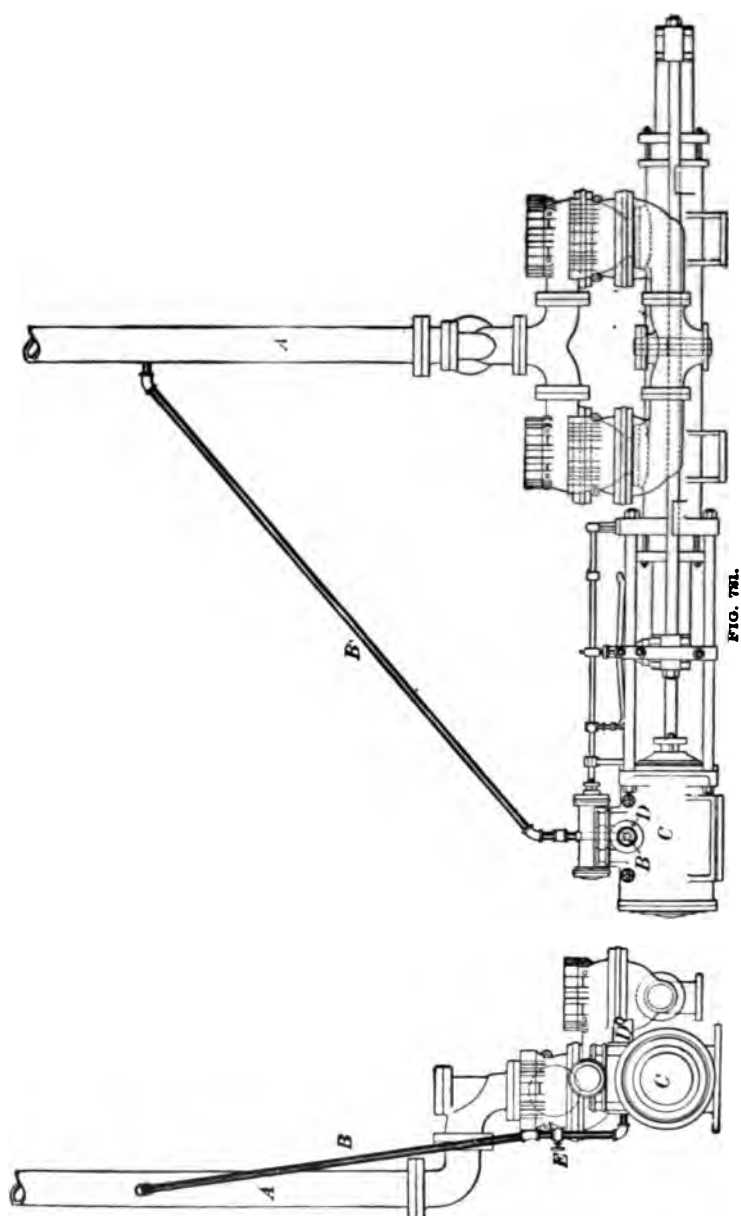
zero.   Any moisture remaining in the air is instantly con-
verted into ice, the exhaust-passages are stopped, and the
engine or pump refuses to work.   This is remedied to some
extent by surrounding the cylinder walls by a warm-water
jacket, the heat being taken from the water and used to
raise the temperature of the expanding air.   Nor is this con-
dition of affairs helped much by using the full air-pressure
throughout the stroke; for the fall in temperature is nearly
as great when it exhausts from the cylinder as it is during
expansion.   In order to use air expansively to any extent,
it must be reheated near or in the cylinder before being
allowed to do work.

2145. Another device for preventing the freezing of
the moisture, and stopping the exhaust-passages through
the accumulation of ice, is shown in Fig. 721. *A* is the de-
livery (column) pipe of a pump operated by compressed air.
B is a small auxiliary pipe leading from *A* to the exhaust-
port of the air-cylinder *C.* This latter pipe enters a short
distance into the exhaust-port directly opposite the exhaust-
pipe *D.* When the valve *E* is opened, a small stream of
water is injected into the exhaust-pipe.   The end of the
pipe *B,* which is connected with the exhaust-port, has a large
number of small holes in it, so that the inflowing water is
delivered in the form of a spray, mixing with the exhaust
air and raising the temperature to such an extent that the
moisture no longer freezes and clogs up the exhaust-passage.
This is a cheap and an easily applied device; it works well
in actual practice.

2146. Before explaining the methods of reheating the
air and the advantages to be derived therefrom, a plant will
be described where electricity is used to operate a com-
pressor near the drills.   A rough sketch of the plant is
shown in Fig. 722. *A* is the motor, which is operated by a
dynamo at the surface.   The current is conveyed down the
shaft and to the motor by means of the wires *E.*   The
motor drives the air-compressor *B.*   *C* is the receiver, which
is felted, so as to retain as much of the heat as possible; the
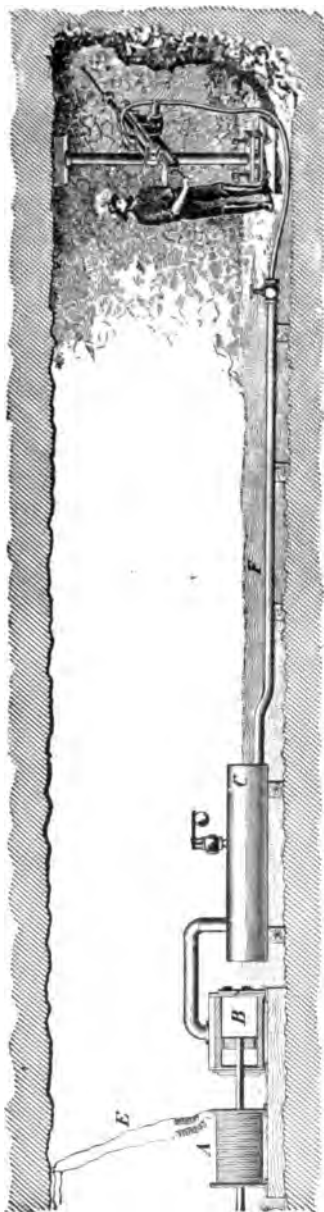
pipe F is also felted for the same reason. Since, in this case, the air-compressor is situated close to the drill, but little heat is lost in transmission. Hence, the air is compressed adiabatically, and as much of the heat is retained as is possible, no effort being made to cool the air during compression. The air thus enters the drill very hot, and leaves it at a temperature but a little lower than that of the free air. Since the electric energy can be conveyed for several miles from the generating station with an efficiency of about 80%, and there is practically no loss due to friction of the air, the efficiency of the whole apparatus is quite high.

2147. The advantages to be gained from heating the air after it has cooled, owing to transmission through great distances, are many. The air enters the cylinder at a very high temperature, and can be used expansively without the danger of freezing its moisture and stopping the exhaust-outlet. The work which can be obtained from a given quantity of compressed air is greatly increased, and the cost of reheating is slight. Unless the reheating is

done in the cylinder of the engine itself, which the compressed air drives, and that during expansion (a scheme which has not yet been realized practically), there will be no increase in pressure.   This is because air expands when heated, and unless prevented from expanding, the pressure will not increase.   The very long column of air behind that which is being reheated acts like an elastic cushion, and the increase of volume is so slight compared with the whole volume in the pipe and receiver that the increase in pressure is not perceptible.   The explanation of the increase of work lies in the fact that all work obtained from air, steam, or gas, when used as a motion-producer, is derived from the amount of heat contained in it.   When the air is heated, almost the entire amount of heat generated by the combustion is converted directly into work, while in the best steam-engines not more than 12% to 13% of the heat energy of the coal is converted into work.

2148.   Fig. 723 shows a reheater in which the air is brought into contact with the fuel.   This illustration is taken from a case which was put into actual service in connection with a rock-drill.   Immediately above the throttle-valve *A*, and near the steam (air) chest *B* of the rock-drill, was placed an enlarged pipe-fitting *C*, in the interior of which, a little above the center, was fixed a piece of wire gauze *D*; above this gauze, charcoal *F* was thrown, some of it being in an incandescent state.   The whole chamber was closed and the compressed air turned on. The air thus brought into direct contact with the burning charcoal was admitted into the drill-cylinder extremely hot.

FIG. 723.
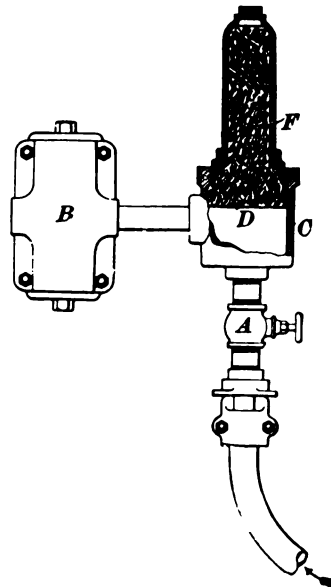
Instead of charcoal, a substance called **sestalit** has been used with considerable success, the advantage of sestalit being that it remains ignited for some length of time after the air has been shut off, and the products of combustion are not objectionable when discharged in a confined space.

2149. Fig. 724 shows an electric reheater. *C* is the air-compressor, the compressed air being conveyed through a pipe to the receiver *D*, and thence by means of the
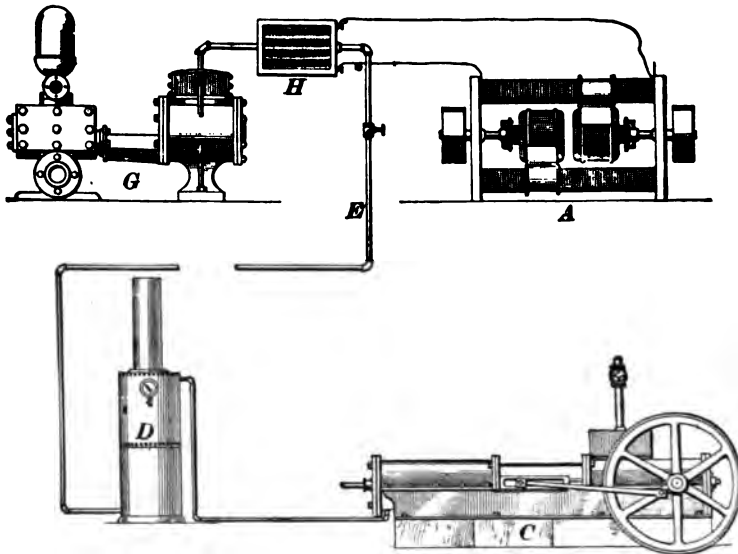
FIG. 724.

pipe *E* to the pump *G*, which is driven by compressed air, and is situated, say, a mile from the compressor. A dynamo *A*, which serves to light the mines, is situated in the engine-room, near the compressor. Near the pump a resistance-coil *H* is placed in a chamber through which the compressed air must pass before entering the pump-cylinder. This resistance-coil is made of some highly refractory substance, which resists the passage of the current to such an extent that the electrical energy is converted into heat, and thus heats the air. This reheater has many advantages. There being no combustion, it is perfectly safe in mines

filled with inflammable gases, and the ease with which it is applied or stopped by simply opening or closing a switch also recommends it.   It is a cheap device, and has been recently employed in the shape of a simple coil of wire placed in the air-pipe.   Since the loss of electric energy is slight compared with the loss of pressure in the compressed air, the efficiency of the whole system is increased by using the electric wire to reheat the air.   It is not, however, as economical as the reheater previously mentioned, but is in many cases more convenient, and for that reason preferable.

Experience has shown that air-engines do not work to advantage at higher temperatures than 350°; hence, the gain through reheating the air is limited.   The cost of the fuel consumed during reheating is trifling.   With the re-heaters commonly used, where the air is heated directly through the combustion of charcoal, it amounts to from one to two cents per horsepower per day.   The gain is considerable, and they should be used when practicable.

THE CALCULATION OF THE SIZE OF AN AIR-COMPRESSOR.

2150.   It is required to determine the size of the steam and air cylinders of a duplex air-compressor to furnish the compressed air necessary to drive a pump and a set of rock-drills, 28 horsepower being necessary to drive them.   To calculate this problem accurately is very tedious and difficult.   Moreover, it requires a good knowledge of the application of logarithms and also of higher mathematics to approach the subject with any degree of success.   Such being the case, the following approximate method will give results close enough for ordinary practice.   The loss of power, in common practice, where compressed air is used to drive machinery in mines and tunnels, is about 70% when common American air-compressors are used and the air is transmitted far enough to lose the heat imparted to it by compression.   Where the best compressors are used, the loss is about 60%.   In both cases, it is assumed that the air

has not been reheated before being used.   If the best com-
pressors and the best reheating systems are used, the loss
may be reduced to about 20%.   Assume for the present case
that the loss is 60%.   Since the horsepower required is 28 and
the loss 60%, the horsepower of the engines must be 28 ÷
.40 = 70 horsepower.   As the engines are to be of the duplex
type, each cylinder must develop 70 ÷ 2 = 35 horsepower.

2151.   To obtain the size of the cylinder, the mean
effective pressure (M. E. P.) must be known.   This may
be obtained in cases like the present, where indicator-cards
can not be taken, from formula **144,** using the constants
obtained from Table **44,** of *Steam-Engines,* Art. **2069.**

2152.   Suppose that the boiler pressure in the present
case is 70 lb., that the engine cuts off at half stroke, and
that it is non-condensing.   The constant, taken from Table
44, for $\frac{1}{2}$ cut-off is .847.   Using formula **144,** M. E. P. =
.9 [.847 (70 + 14.7) − 17] = 49.27 lb. per sq. in.   The diam-
eter of the steam-cylinder may be calculated by the following
formula:

$$D = 79.6 \sqrt[3]{\frac{H}{r\,PN}}, \qquad \textbf{(148.)}$$

in which $H =$ the number of horsepower the engine is to
                develop;
         $D =$ diameter of cylinder in inches;
         $r =$ ratio of length of stroke to diameter of cylinder;
         $P =$ mean effective pressure per sq. in. on the piston;
         $N =$ number of strokes per minute.

In the present case, assume that the stroke is $1\frac{1}{4}$ times
the diameter; then, $r = 1\frac{1}{4}$.   Also, that the number of strokes
per   minute = 300;   then, $D = 79.6 \sqrt[3]{\dfrac{35}{1\frac{1}{4} \times 49.27 \times 300}} =$
9.849 inches, or say $9\frac{7}{8}$ in.   Length of stroke = $9\frac{7}{8} \times 1\frac{1}{4} =$
12.344 inches, or say $12\frac{3}{8}$ inches.   If the air is to be com-
pressed to about 60 or 70 lb., it is good practice to make the
air-cylinders the same in size as the steam-cylinders.   In
the present example, it would be good practice to make all
four cylinders 10 by 12 inches.

PHYSICAL PROPERTIES OF AIR AND GASES.

2153. Air is a mechanical mixture of two gases, nitrogen and oxygen, and contains about three parts, by weight, of the former, to one of the latter. As *water* is the most common type of fluids, so *air* is the most common type of gases. It was supposed by the ancients that air had no weight, and it was not until about the year 1650 that it was proven that air really has weight. A cubic inch of air, under ordinary conditions, weighs .31 grain, nearly. The ratio of the weight of air to water is about 1 : 774; that is, air is only $\frac{1}{774}$ as heavy as water. If a vessel made of light material be filled with a gas lighter than air, so that the total weight of the vessel and gas is less than the air they displace, the vessel will rise. It is on this principle that balloons are made.

2154. Since air has weight, it is evident that the enormous quantity of air that constitutes the atmosphere must exert a considerable pressure upon the earth. This is easily proven by taking a long glass tube closed at one end and filling it with mercury. If the finger be placed over the open end so as to keep the mercury from running out, and the tube be inverted and placed in a glass of mercury, as shown in Fig. 725, the mercury in the tube will fall, then rise, and, after a few oscillations, will come to rest at a height above the top of the mercury in the glass equal to about 30 inches. This

Fig. 725.

24—17
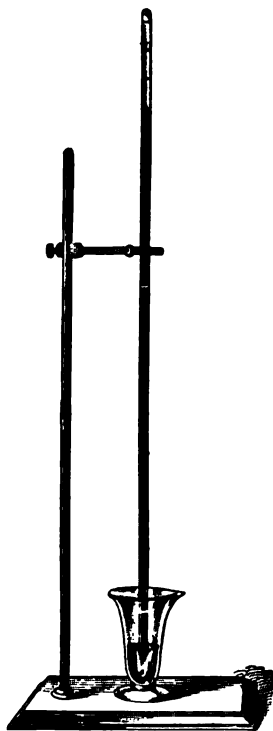
height will always be the same under the same atmospheric conditions. Now, if the atmosphere has weight, it must press upon the upper surface of the mercury in the glass with equal intensity upon every square unit, except upon that part of the surface occupied by the tube. In order that there will be equilibrium, the weight of the mercury in the tube must be equal to the pressure of the air upon an area of the upper surface of the mercury in the glass equal to the area of the inside of the tube. Suppose that the area of the inside of the tube is 1 square inch; then, since mercury is 13.6 times as heavy as water, and a cubic inch of water weighs .03617 pound, the weight of the mercurial column is .03617 × 13.6 × 30 = 14.7574 pounds. The actual height of the mercury is a little less than 30 inches, and the actual weight of a cubic inch of distilled water is a little less than .03617 pound. When these considerations are taken into account, the average weight of the mercurial column at the level of the sea, when the temperature is 60°, is 14.69 pounds, or, practically, 14.7 pounds. Since this weight, exerted upon 1 square inch of the liquid in the glass, just produced equilibrium, it is plain that the pressure of the outside air is 14.7 pounds upon every square inch of surface.

2155. Vacuum.—The space between the upper end of the tube and the upper surface of the mercury is called a *Toricellian vacuum*, or simply a *vacuum*, meaning that it is an entirely empty space, and does not contain any substance, solid, liquid, or gaseous. If there was a gas of some kind there, no matter how small the quantity might be, it would expand, filling the space, and its tension would cause the column of mercury to fall and become shorter, according to the amount of gas or air present. The space is then called a *partial vacuum*. If the mercury fell 1 inch, so that the column was only 29 inches high, we would say, in ordinary language, that there were *29 inches of vacuum*. If it fell 8 inches, we would say that there were 22 inches of vacuum; if it fell 16 inches, we would say that there

were 14 inches of vacuum, etc.   Hence, when the vacuum‑
gauge of a condensing-engine shows 26 inches of vacuum,
there is enough air in the condenser to produce
a pressure of $\dfrac{30 - 26}{30} \times 14.7 = \dfrac{4}{30} \times 14.7 = 1.96$
pounds per square inch.

If the tube had been filled with water instead
of mercury, the height of the column of water to
balance the pressure of the atmosphere would
have been $30 \times 13.6 = 408$ inches $= 34$ feet.   This
means that if a tube be filled with water, inverted,
and placed in a dish of water in a manner similar
to the experiment made with the mercury, the
height of the column of water would be 34 feet.

2156.   The **barometer** is an instrument
used for measuring the pressure of the atmosphere.
There are two kinds in general use, the mercu‑
rial barometer and the aneroid barometer.   The
mercurial barometer is shown in Fig. 726.   The
principle is the same as the inverted tube,
shown in Fig. 725.   In this case, the tube and
cup at the bottom are protected by a brass or iron
casing.   Near the top of the tube is a graduated
scale which can be read to $\frac{1}{1000}$ of an inch by
means of a vernier.   Attached to the casing is
an accurate thermometer for determining the
temperature of the outside air at the time the
barometric observation is taken.   This is neces‑
sary, since mercury expands when the temperature
is increased, and contracts when the temperature
falls; for this reason, a standard temperature is as‑
sumed, and all barometer readings are reduced to

FIG. 726.

this temperature.   This standard temperature is
usually taken at 32° F., at which temperature the height of
the mercurial column is 30 inches.   Another correction is
made for the altitude of the place above sea-level, and a
third correction for the effects of capillary attraction.

In Fig. 727 is shown a cut of an *aneroid barometer*. These instruments are made in various sizes, from the size of a watch up to an 8 or 10 inch face. They consist of a cylindrical box of metal, with a top of thin, elastic, corrugated metal. The air is removed from the box. When the atmospheric pressure increases, the top is pressed inwards, and, when it is diminished, the top is pressed outwards by its

FIG. 727.

own elasticity, aided by a spring beneath. These movements of the cover are transmitted and multiplied by a combination of delicate levers, which act upon an index-hand, and cause it to move either to the right or left, over a graduated scale. These barometers are self-correcting (compensated) for variations in temperature. They are very portable, occupying but a small space, and are so delicate that they are said to show a difference in the atmospheric pressure when transferred from the table to the

. The mercurial barometer is the standard. With
is with water, the lower we get, the greater the pressure,
he higher we get, the less the pressure. At the level
e sea, the height of the mercurial column is about 30
s; at 5,000 feet above the sea, it is 24.7 inches; at
0 feet above the sea, it is 20.5 inches; at 15,000 feet, it
9 inches; at 3 miles, it is 16.4 inches, and at 6 miles
e the sea-level, it is 8.9 inches.

57. Density of Air.—The weight of a cubic foot
d the **density**) also varies with the altitude; that is, a
: foot of air at an elevation of 5,000 feet above the
vel will not weigh as much as a cubic foot at sea-level.
is proven conclusively by the fact that at a height of
iles the mercurial column measures but 15 inches, in-
ing that half the weight of the entire atmosphere is
r that. It is known that the height of the earth's at-
here is at least 50 miles; hence, the air just before
ing the limit must be in an exceedingly rarefied state.
y means of barometers that great heights are measured.
ineroid barometer has the heights marked on the dial,
at they can be read directly. With the mercurial ba-
ter, the heights must be calculated from the reading.

58. Atmospheric Pressure.—The atmospheric
ure is everywhere present, and presses all objects in all
tions with equal force. If a book is laid upon the
, the air presses upon it in every direction with an equal
ge force of 14.7 pounds per square inch. It would
as though it would take considerable force to raise a
from the table, since, if the size of the book were 8
s by 5 inches, the pressure upon it is $8 \times 5 \times 14.7 =$
ounds; but there is an equal pressure beneath the book
unteract the pressure on the top. It would now seem
ough it would require a great force to open the book,
there are two pressures of 588 pounds each, acting in
ite directions, and tending to crush the book; so it
l but for the fact that there is a layer of air between
leaf acting upwards and downwards with a pressure of

14.7 pounds per square inch.   If two metal plates be made
as perfectly smooth and flat as it is possible to get them,
and the edge of one be laid upon the edge of the other, so
that one may be slid upon the other, and thus exclude the air,
it will take an immense force, compared with the weight of
the plates, to separate them.   This is because the full pres-
sure of 14.7 pounds per square inch is then exerted upon each
plate, with no counteracting equal pressure between them.

If a piece of flat glass be laid upon a flat surface that has
been previously moistened with water, it will require con-
siderable force to separate them; this is because the water
helps to fill up the pores in the flat surface and glass, and
thus creates a partial vacuum between the glass and the sur-
face, thereby reducing the counter-pressure beneath the glass.

2159.   Tension of Gases.—In Fig. 725, the space
above the column of mercury was said to be a vacuum, and
that if any gas or air was present it would expand, its ten-
sion forcing the column of mercury downwards.   If enough
gas is admitted to cause the mercury to stand at 15 inches,
the tension of the gas is evidently $\dfrac{14.7}{2} = 7.35$ pounds per
square inch, since the pressure of the outside air of 14.7
pounds per square inch balances only 15 inches, instead of 30
inches of mercury; that is, it balances only half as much as
it would if there were no gas in the tube; therefore, the
tension (pressure) of the gas in the tube is 7.35 pounds.
If more gas is admitted, until the top of the mercurial col-
umn is just level with the mercury in the cup, the gas in
the tube has then a tension equal to the outside pressure of
the atmosphere.   Suppose that the bottom of the tube is
fitted with a piston, and that the total length of the inside
of the tube is 36 inches.   If the piston be shoved upwards
so that the space occupied by the gas is 18 inches long, in-
stead of 36 inches, the temperature remaining the same as
before, it will be found that the tension of the gas within
the tube is 29.4 pounds.   It will be noticed that the volume
occupied by the gas is only half that in the tube before the

ı was moved, while the pressure is twice as great, since
‹ 2 = 29.4 pounds. If the piston be shoved up so
:he space occupied by the gas is only 9 inches instead
inches, the temperature still remaining the same, the
ıre will be found to be 58.8 pounds per square inch.
volume has again been reduced one-half, and the
ıre increased two times, since 29.4 × 2 = 58.8 pounds.
volume now occupied by the gas is 9 inches long,
:as, before the piston was moved, it was 36 inches long;
: tube was assumed to be of uniform diameter through-
ı length, the volume is now $\frac{9}{36}$ = $\frac{1}{4}$ of its original volume,
its pressure is $\frac{58.8}{14.7}$ = 4 times its original pressure.

ıver, if the temperature of the confined gas remains
ıme, the pressure and volume will always vary in a
r way. The law which states these effects is called
ıtte's law.

60. Marlotte's Law.—*The temperature remaining
ıme, the volume of a given quantity of gas varies in-
v as the pressure.*

: meaning of this is: If the volume of the gas is dimin-
to $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{5}$, etc., of its former volume, the tension will
:reased 2, 3, 5, etc., times, or, if the outside pressure
reased 2, 3, 5, etc., times, the volume of the gas will be
ished to $\frac{1}{2}$, $\frac{1}{3}$, $\frac{1}{5}$, etc., of its original volume, the tem-
ıre remaining constant. It also means that if a gas
.er a certain pressure, and the pressure is diminished
$\frac{1}{2}$, $\frac{1}{10}$, etc., of its original pressure, that the volume of
ınfined gas will be increased 2, 4, 10, etc., times, its
n decreasing at the same rate.

ıpose 3 cubic feet of air to be under a pressure of 60
ls per square inch in a cylinder fitted with a movable
ı; then, the product of the volume and pressure is 3 ×
180. Let the volume be increased to 6 cubic feet; then,
·essure will be 30 pounds per square inch, and 30 ×
30 as before. Let the volume be increased to 24 cubic
it is then $\frac{24}{3}$ = 8 times its original volume, and the

pressure is $\frac{1}{8}$ of its original pressure, or $60 \times \frac{1}{8} = 7\frac{1}{2}$ pounds, and $24 \times 7\frac{1}{2} = 180$, as in the two preceding cases. It will now be noticed that if a gas be enclosed within a confined space, and allowed to expand without losing any heat, *the product of the pressure and the corresponding volume for one position of the piston is the same as for any other position of the piston.* If the piston was to compress the air, the rule would still hold good.

Let p = pressure for one position of the piston;
p_1 = pressure for any other position of the piston;
v = volume corresponding to the pressure p;
v_1 = volume corresponding to the pressure p_1.

Then, $p\,v = p_1\,v_1;$ **(149.)**

also, $p_1 = \dfrac{p\,v}{v_1};$ **(150.)**

and $v_. = \dfrac{p\,v}{p_1}.$ **(151.)**

Knowing the volume and the pressure for any position of the piston and the volume for any other position, the pressure may be calculated by formula **150,** or if the pressure is known for any other position, the volume may be calculated by formula **151.**

EXAMPLE.—If 1.875 cubic feet of air be under a pressure of 72 pounds per square inch, (*a*) what will be the pressure when the volume is increased to 2 cubic feet? (*b*) to 3 cubic feet? (*c*) to 9 cubic feet?

SOLUTION.—(*a*) $p_1 = \dfrac{p\,v}{v_1} = \dfrac{72 \times 1.875}{2} = 67\frac{1}{2}$ pounds per square inch. Ans.

(*b*) $p_1 = \dfrac{72 \times 1.875}{3} = 45$ pounds per square inch. Ans.

(*c*) $p_1 = \dfrac{72 \times 1.875}{9} = 15$ pounds per square inch. Ans.

EXAMPLE.—If 10 cubic feet of air have a tension of 5.6 pounds per square inch, (*a*) what is the volume when the tension is 4 pounds? (*b*) 8 pounds? (*c*) 25 pounds? (*d*) 100 pounds?

Solution.—(a) $v_1 = \dfrac{p\,v}{p_1} = \dfrac{5.6 \times 10}{4} = 14$ cubic feet. **Ans.**

(b) $v_1 = \dfrac{5.6 \times 10}{8} = 7$ cubic feet. **Ans.**

(c) $v_1 = \dfrac{5.6 \times 10}{25} = 2.24$ cubic feet. **Ans.**

(d) $v_1 = \dfrac{5.6 \times 10}{100} = .56$ cubic foot. **Ans.**

2161. As a necessary consequence of Mariotte's law, it may be stated that *the density of a gas varies directly as the pressure and inversely as the volume;* that is, *the density increases as the pressure increases, and decreases as the volume increases.*

This is evident, since if a gas has a tension of two atmospheres, or $14.7 \times 2 = 29.4$ pounds per square inch, it will weigh twice as much as the same volume would if the tension was one atmosphere, or 14.7 pounds per square inch. For, let the volume be increased until it is twice as great as · the original volume, the tension will then be one atmosphere. The total weight of the gas has not been changed, but there are now 2 cubic feet for every 1 cubic foot of the original volume, and the weight of 1 cubic foot now is only half as great as before. Thus, the density decreases as the volume increases, and as an increase of pressure causes a decrease of volume, the density increases as the pressure increases.

Let D be the density corresponding to the pressure p and volume v, and D_1 be the density corresponding to the pressure p_1 and volume v_1. Then,

$$ p : D :: p_1 : D_1, \text{ or } p\,D_1 = p_1\,D, \qquad \textbf{(152.)} $$

and $\qquad v : D_1 :: v_1 : D, \text{ or } v\,D = v_1\,D_1. \qquad \textbf{(153.)}$

2162. Since the weight is proportional to the density, the weights may be used in place of the densities in formulas **152** and **153.** Thus, let W be the weight of a quantity of air or other gas whose volume is v and pressure is p; let W_1 be the weight of the same quantity when the volume is v_1 and pressure is p_1. Then,

$$ p : W :: p_1 : W_1, \text{ or } p\,W_1 = p_1\,W; \qquad \textbf{(154.)} $$

$$ v : W_1 :: v_1 : W, \text{ or } v\,W = v_1\,W_1. \qquad \textbf{(155.)} $$

EXAMPLE.—The weight of 1 cubic foot of air at a temperature of 60° F. and under a pressure of 1 atmosphere (14.7 pounds per square inch) is .0763 pound; what would be the weight per cubic foot if the volume was compressed until the tension was 5 atmospheres, the temperature still being 60° ?

SOLUTION.—Using formula **154**,

$p : W :: p_1 : W_1$, or $1 : .0763 :: 5 : W_1$, or $W_1 = .3815$ lb. Ans.

EXAMPLE.—If in the last example the air had expanded until the tension was 5 pounds per square inch, what would have been its weight per cubic foot ?

SOLUTION.—Here $p = 14.7$, $p_1 = 5$, and $W = .0763$. Hence, using the same formula, $14.7 : .0763 :: 5 : W_1$, or $W_1 = .02595$ lb. Ans.

EXAMPLE.—If 6.75 cubic feet of air at a temperature of 60° F., and a pressure of one atmosphere, are compressed to 2.25 cubic feet (the temperature still remaining 60° F.), what is the weight of a cubic foot of the compressed air ?

SOLUTION.—Using formula **155**,

$$v : W_1 :: v_1 : W, \text{ or } 6.75 : W_1 :: 2.25 : .0763$$

or
$$W_1 = \frac{.0763 \times 6.75}{2.25} = .2289 \text{ lb. Ans.}$$

2163. Relation of Temperature to Volume.—In all that has been said before, it has been stated that the temperature was constant; the reason for this will now be explained. Suppose 5 cubic feet of air to be confined in a cylinder whose area is 10 square inches, placed in a vacuum so that there will be no pressure due to the atmosphere, and the cylinder be fitted with a piston weighing say 100 pounds. The tension of the gas will be $\frac{100}{10} = 10$ pounds per square inch. Suppose that the temperature of the air is 32° F., and that it is heated until the temperature is 33° F., or the temperature is increased 1°; it will be found that the piston has risen a certain amount, and, consequently, the volume has increased, while the pressure is the same as before, or 10 pounds per square inch. If more heat is applied until the temperature of the gas is 34° F., it will be found that the piston has again risen and the volume again increased, while the pressure still remains the same. It will be found that for every increase of temperature there will be a corresponding increase of volume. The law which expresses this change is called *Gay-Lussac's law*.

2164. Gay-Lussac's Law.—*If the pressure remains constant, every increase of temperature of 1° F. produces in a given quantity of gas an expansion of $\frac{1}{4\sqrt{1}}$ of its volume at 32° F.*

If the pressure remains constant, it will also be found that every decrease of temperature of 1° F. will cause a decrease of $\frac{1}{4\sqrt{1}}$ of the volume at 32° F.

Let v = volume of gas before heating;
  v_1 = volume of gas after heating;
  t = temperature corresponding to volume v;
  t_1 = temperature corresponding to volume v_1.

Then, $$v_1 = v\left(\frac{459 + t_1}{459 + t}\right). \qquad \textbf{(156.)}$$

That is, *the volume of gas after heating (or cooling) equals the original volume multiplied by 459 plus the final temperature, divided by 459 plus the original temperature.*

EXAMPLE.—When 5 cubic feet of air at a temperature of 45° are heated under constant pressure up to 177°, what is its new volume?

SOLUTION.— $v_1 = v\left(\frac{459 + t_1}{459 + t}\right) = 5 \times \left(\frac{636}{504}\right) = 6.309$ cu. ft.   Ans.

Suppose that a certain volume of gas is confined in a vessel so that it can not expand; in other words, suppose that the piston of the cylinder before mentioned to be fastened so that it can not move. Let a gauge be placed on the cylinder so that the tension of the confined gas can be registered. If the gas is heated, it will be found that for every increase of temperature of 1° F. there will be a corresponding increase of $\frac{1}{4\sqrt{1}}$ of the tension at 32° F.; that is, the volume remaining constant, the tension increases $\frac{1}{4\sqrt{1}}$ of the tension at 32° F. for every degree rise of temperature.

Let p = the original tension;
  t = the corresponding temperature;
  t_1 = any higher temperature;
  p_1 = corresponding tension.

Then, $$p_1 = p\left(\frac{459 + t_1}{459 + t}\right). \qquad \textbf{(157.)}$$

That is, *if a certain quantity of gas is heated from $t°$ $t_1°$, the volume remaining constant, the resulting tension p will be equal to the original tension multiplied by 459 plus the final temperature, divided by 459 plus the original temperature.*

EXAMPLE.—If a certain quantity of air is heated under constant volume from 45° to 177°, what is the resulting tension, the original tension being 14.7 pounds per square inch ?

SOLUTION.— $p_1 = p\left(\dfrac{459 + t_1}{459 + t}\right) = 14.7 \times \left(\dfrac{636}{504}\right) = 18.55$ lb. per sq. in. Ans.

2165. Absolute Zero.—According to the modern and now generally accepted theory of heat, the atoms and molecules of all bodies are in an incessant state of vibration. The vibratory movement in the liquids is faster than in the solids; it is faster in the gases than in either of the others. Any increase of heat increases the vibrations, and a decrease of heat decreases them. From experiments and calculations based upon higher mathematics, it has been concluded that at 459° below zero on the Fahrenheit scale, or at 273° below zero on the Centigrade scale, all these vibrations cease. This point is called the *absolute zero*, and all temperatures reckoned from this point are called the *absolute temperatures*. The point of absolute zero has never been reached nor closely approached, the lowest recorded temperature being 360° F. below zero, but, nevertheless, it has a meaning, and is used in many formulas, being nearly always denoted by T. The ordinary temperatures are denoted by t. When the word temperature alone is used, the meaning is the same as ordinarily used, but when absolute temperature is specified, 459° F. must be added to the temperature. The absolute temperature corresponding to 212° F. is 459° + 212° = 671° F. If the absolute temperature is given, the ordinary temperature may be found by subtracting 459° from the absolute temperature. Thus, the absolute temperature being 520° F., what is the temperature ?

$$520° - 459° = 61° \text{ F.}$$

Let $P=$ pressure of air per square inch;
$V=$ volume of air in cubic feet;
$T=$ absolute temperature of air;
$W=$ weight of air in pounds.

Then,

$$P = \frac{.37052\ W\,T}{V};\qquad \textbf{(158.)}$$

$$V = \frac{.37052\ W\,T}{P};\qquad \textbf{(159.)}$$

$$T = \frac{P\,V}{.37052\ W};\qquad \textbf{(160.)}$$

$$W = \frac{P\,V}{.37052\ T}.\qquad \textbf{(161.)}$$

EXAMPLE.—If 40 cubic feet of air weigh 3.5 pounds, and have a temperature of 82°, what is the pressure (tension) in pounds per square inch?

SOLUTION.— $P = \dfrac{.37052\ W\,T}{V} = \dfrac{.37052 \times 3.5 \times 541}{40} = 17.539$ lb. per sq. in. Ans.

EXAMPLE.—What is the volume in cubic feet of a certain quantity of air having a tension of 17.539 pounds per square inch, a temperature of 80°, and which weighs 3.5 pounds?

SOLUTION.— $V = \dfrac{.37052\ W\,T}{P} = \dfrac{.37052 \times 3.5 \times 541}{17.539} = 40$ cu. ft. Ans.

EXAMPLE.—If 40 cubic feet of air having a tension of 17.539 pounds per square inch weigh 3.5 pounds, what is the temperature?

SOLUTION.— $T = \dfrac{P\,V}{.37052\ W} = \dfrac{17.539 \times 40}{.37052 \times 3.5} = 541°$, nearly.   Hence, $541° - 459° = 82°$.   Ans.

EXAMPLE.—If 40 cubic feet of air have a tension of 17.539 pounds per square inch, and a temperature of 82°, (*a*) what is its weight? (*b*) what is its weight per cubic foot?

SOLUTION.—(*a*) $W = \dfrac{P\,V}{.37052\ T} = \dfrac{17.539 \times 40}{.37052 \times 541} = 3.5$ lb.   Ans.

(*b*) $3.5 \div 40 = .0875$ lb. per cu. ft.   Ans.

2166.   Mixing of Gases.—If two liquids which do not act chemically upon each other are mixed together and allowed to stand, it will be found that after a time the two

liquids have separated, and the heavier has fallen to the
bottom. If two equal vessels containing gases of different
densities be put in communication with each other, they
will be found to have mixed in equal proportions after a
short time. If one vessel be above the other, and the
heavier gas be in the lower vessel, the same result will occur.
The greater the difference of the densities of the two gases,
the quicker they will mix. It is assumed that no chemical
action takes place between the two gases. When the two
gases have the same temperature and pressure, the pressure
of the mixture will be the same; this is evident, since the
total volume has not been changed, and unless the volume
or temperature changes, the pressure can not change. This
property of the mixing of gases is a very valuable one, since,
if they acted like liquids, carbonic acid gas (the result of
combustion), which is $1\frac{1}{2}$ times as heavy as air, would remain
next to the earth, instead of dispersing into the atmosphere,
the result being that no animal life could exist.

**2167. Mixtures of Equal Volumes of Gases
Having Unequal Pressures.**—*If two gases having the
same volume and temperature, but different pressures, be mixed
in a vessel whose volume equals one of the equal volumes of
the gas, the pressure of the mixture will be equal to the sum
of the two pressures, provided that the temperature remains
the same as before.*

EXAMPLE.—Two vessels containing 3 cubic feet of gas, each at a
temperature of 60°, and at a pressure of 40 pounds and 25 pounds per
square inch, respectively, are placed in communication with each other,
and all the gas is compressed into one vessel. If the temperature of
the mixture is also 60°, what is the pressure?

SOLUTION.—According to the rule just given, the pressure will be
$40 + 25 = 65$ lb. per sq. in. Ans.

**2168. Mixture of Two Gases Having Unequal
Volumes and Pressures.**—

Let v and p be the volume and pressure of one of the gases.
Let v_1 and p_1 be the volume and pressure of the other gas.
Let V and P be the volume and pressure of the mixture.

Then, if the temperature remains the same,

$$P = \frac{p\,v + p_1\,v_1}{V}; \quad \textbf{(162.)}$$

$$V = \frac{p\,v + p_1\,v_1}{P}. \quad \textbf{(163.)}$$

EXAMPLE.—Two gases of the same temperature, having volumes of 7 cubic feet and 4½ cubic feet, and tensions of 25 pounds and 18 pounds per square inch, respectively, are mixed together in a vessel whose volume is 10 cubic feet. The temperature remaining the same, what is the resulting pressure?

SOLUTION.— $P = \frac{p\,v + p_1\,v_1}{V} = \frac{(25 \times 7) + (18 \times 4\frac{1}{2})}{10} = \frac{256}{10} = 25.6$ lb. per sq. in. Ans.

EXAMPLE.—What must be the volume of a vessel which will hold two gases whose volumes are 7 cubic feet and 4½ cubic feet, and whose tensions are 25 pounds and 18 pounds per square inch, respectively, in order that the pressure may be 25.6 pounds per square inch, the temperature remaining the same throughout?

SOLUTION.— $V = \frac{p\,v + p_1\,v_1}{P} = \frac{(25 \times 7) + (18 \times 4\frac{1}{2})}{25.6} = 10$ cu. ft. Ans

DROMECHANICS AND PUMPING.

HYDROSTATICS.

LAWS OF LIQUID PRESSURE.

9. Hydrostatics treats of liquids at rest under the
of forces. Liquids are very nearly *incompressible*. A
e of 15 pounds per square inch compresses water less
$\frac{1}{0000}$ of its volume.

0. Fig. 728 represents two cylindrical vessels of ex-
ie same size. The vessel *a* is fitted with a wooden
f the same size as the cyl-
and can move in it; the
b is filled with water,
depth is the same as the
of the wooden block in *a*.
essels are fitted with air-
istons *P*, whose areas are
sq. in.
ose, for convenience, that
ights of the cylinders,
block, and water be
ed, and that a force of
unds be applied to both
The pressure per square
ill be $\frac{100}{10} = 10$ pounds.

FIG. 728.

essel *a*, this pressure will be transmitted to the bottom
vessel, and will be 10 pounds per square inch; it is
see that there will be no pressure on the sides. In
sel *b*, an entirely different result is obtained. The
e on the bottom will be the same as in the other case,

§ 21
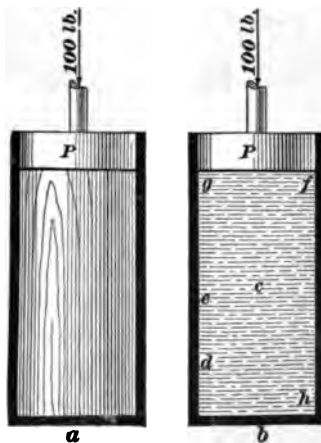
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24—18

that is, 10 pounds per square inch, but, owing to the fact that the molecules of the water are perfectly free to move, this pressure of 10 pounds per square inch is *transmitted in every direction with the same intensity;* that is to say, the pressure at any point, *c, d, e, f, g, h,* etc., due to the force of 100 pounds, is exactly the same, and equals 10 pounds per square inch.

This may be easily proven experimentally by means of an apparatus like that shown in Fig. 729. Let the area of
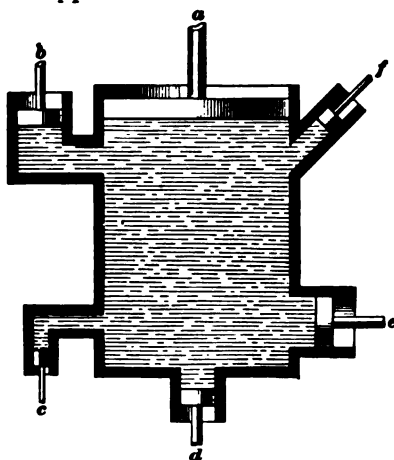
Fig. 729.

the pistons *a, b, c, d, e,* and *f* be 20, 7, 1, 6, 8, and 4 sq. in., respectively.

If the pressure due to the weight of the water be neglected, and a force of 5 pounds be applied at *c* (whose area is 1 sq. in.), a pressure of 5 pounds per square inch will be transmitted in all directions; in order that there shall be no movement, a force of 6 × 5 = 30 pounds must be applied at *d*, 40 pounds at *e*, 20 pounds at *f*, 100 pounds at *a*, and 35 pounds at *b*.

If a force of 99 pounds were applied to *a*, instead of 100 pounds, the piston *a* would rise, and the other pistons *b, c, d, e,* and *f* would move inwards; but, if the force applied to *a* were 100 pounds, they would all be in equilibrium. Suppose 101 pounds to be applied at *a*; the pressure per square inch would be $\frac{101}{20} = 5.05$ pounds, which would be transmitted in all directions; and, since the pressure due to *c* is only 5 pounds per square inch, it is now evident that the piston *a* will move downwards, and the pistons *b, c, d, e,* and *f* will be forced outwards.

2171. The whole may be summed up as follows :

The pressure per unit of area exerted anywhere upon a mass of liquid is transmitted undiminished in all directions, and

*acts with the same intensity upon all surfaces in a direction at **right angles** to those surfaces.*

This law was first discovered by Pascal, and is the most important in hydromechanics. Its meaning should be thoroughly understood.

EXAMPLE.—If the area of the piston *e* in Fig. 729 is 8.25 sq. in., and a force of 150 pounds is applied to it, what forces must be applied to the other pistons to keep the water in equilibrium, assuming that their areas were the same as given before ?

SOLUTION.— $\dfrac{150}{8.25} = 18.182$ pounds per square inch, nearly.

$$20 \times 18.182 = 363.64 \ \text{lb.} = \text{force to balance } a.$$
$$7 \times 18.182 = 127.274 \ \text{lb.} = \text{force to balance } b.$$
$$1 \times 18.182 = \ \ 18.182 \ \text{lb.} = \text{force to balance } c.$$
$$6 \times 18.182 = 109.092 \ \text{lb.} = \text{force to balance } d.$$
$$4 \times 18.182 = \ \ 72.728 \ \text{lb.} = \text{force to balance } f.$$

⎫ Ans.

The pressure due to the weight of a liquid may be downwards, upwards, or sideways.

2172. Downward Pressure.—In Fig. 730 the pressure on the bottom of the vessel *a* is, of course, equal to the weight of the water it contains. If the area of the bottom of the vessel *b* and the depth of the liquid contained in it are the same as in the vessel *a*, the pressure on the bottom of *b* will be the same as on the bottom of *a*. Suppose the bottoms of the vessels *a* and *b* are 6 inches square, that the part *c d*, in the vessel *b*, is 2 inches square, and that the vessels are filled with water. The weight of 1 cubic
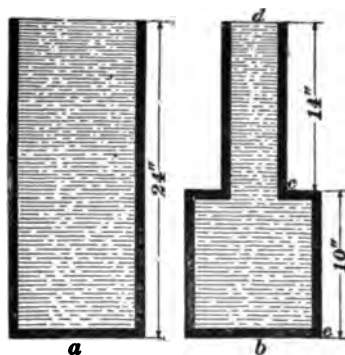
FIG. 730.

inch of water is $\dfrac{62.5}{1,728} = .03617$ pound. The number of cubic inches in $a = 6 \times 6 \times 24 = 864$ cubic inches. The weight of the water is $864 \times .03617 = 31.25$ pounds. Hence, the total pressure on the bottom of the vessel *a* is 31.25 pounds, or

0.868 pound per square inch. The pressure in *b*, due to the
weight contained in the part *b c*, is 6 × 6 × 10 × .03617 =
13.02 pounds. The weight of the part contained in *c d* is
2 × 2 × 14 × .03617 = 2.0255 pounds, and the weight per
square inch of area in *c d* is $\dfrac{2.0255}{4}$ = .5064 pound.

According to Pascal's law, this weight (pressure) is trans-
mitted equally in all directions; therefore, every square inch
of the top of the large part of the vessel *b* will be subjected
to a pressure of .5064 pound. The area of the part *b c* is
6 × 6 = 36 sq. in., and the total pressure due to the weight
of the water in the small part will be .5064 × 36 = 18.23
pounds. Hence, the total pressure on the bottom of *b* will
be 13.02 + 18.23 = 31.25 pounds, the same result as in the
case of the vessel *a*.

If an additional pressure of 10 pounds per square inch
were applied to the upper surface of both vessels, the total
pressure on their bottoms would be 31.25 + (6 × 6 × 10) =
31.25 + 360 = 391.25 pounds.

If in this case this pressure were obtained by means of a
weight placed on a piston, as shown in Figs. 728 and 729, the
weight for the vessel *a* would be 6 × 6 × 10 = 360 pounds,
and for the vessel *b*, 2 × 2 × 10 = 40 pounds.

2173. The general law for the downward pressure
upon the bottom of any vessel:

*The pressure upon the bottom of a vessel
containing a fluid is independent of the shape
of the vessel, and is equal to the weight of
a prism of the fluid whose base is the same as
the bottom of the vessel, and whose altitude is
the distance between the bottom and the upper
surface of the fluid, plus the pressure per
unit of area upon the upper surface of the
fluid, multiplied by the area of the bottom of
the vessel.*

FIG. 731.   Suppose that the vessel *b*, Fig. 730, were
inverted, as shown in Fig. 731; the pressure upon the bottom

will still be 0.868 pound per square inch, but it will require a weight of 3,490 pounds to be placed upon a piston at the upper surface to make the pressure on the bottom 391.25 pounds, instead of a weight of 40 pounds, as in the other case.

EXAMPLE.—A vessel filled with salt water, having a specific gravity of 1.03, has a circular bottom 13 inches in diameter. The top of the vessel is fitted with a piston 3 inches in diameter, on which is laid a weight of 75 pounds. What is the total pressure on the bottom, if the depth of the water is 18 inches?

SOLUTION.—The weight of 1 cubic inch of the water is $\dfrac{62.5 \times 1.03}{1,728} =$ .037254 lb. $13 \times 13 \times .7854 \times 18 \times .037254 = 89.01$ pounds = the pressure due to the weight of the water. $\dfrac{75}{3 \times 3 \times .7854} = 10.61$ pounds per square inch due to the weight on the piston. $13 \times 13 \times .7854 \times 10.61 = 1,408.29$ pounds.

Total pressure $= 1,408.29 + 89.01 = 1,497.3$ pounds. Ans.

2174. Upward Pressure.—In Fig. 732 is represented a vessel of exactly the same size as that represented in Fig. 731. There is no upward pressure on the surface c due to the weight of the water in the large part $c\,d$, but there is an upward pressure on c due to the weight of the water in the small part $b\,c$. The pressure per square inch due to the weight of the water in $b\,c$ was found to be .5064 pound (see Art. **2172**); the area of the upper surface c of the large part $c\,d$ is evidently $(6 \times 6) - (2 \times 2) = 36 - 4 = 32$ sq. in., and the total upward pressure due to the weight of the water is $.5064 \times 32 = 16.2$ pounds.
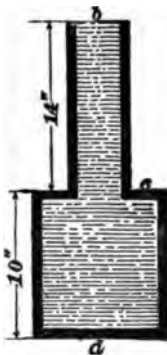
FIG. 732.

If an additional pressure of 10 pounds per square inch were applied to a piston fitting the top of the vessel, the total upward pressure on the surface c would be

$$16.2 + (32 \times 10) = 336.2 \text{ pounds.}$$

2175. General law for upward pressure:

The upward pressure on any submerged horizontal surface equals the weight of a prism of the liquid, whose base has an area equal to the area of the submerged surface, and whose

altitude is the distance between the submerged surface ana the upper surface of the liquid, plus the pressure per unit oj area on the upper surface of the fluid, multiplied by the area of the submerged surface.

EXAMPLE.—A horizontal surface, 6 inches by 4 inches, is submerge in a vessel of water 26 inches below the upper surface. If the pressur on the water is 16 pounds per square inch, what is the total upwar pressure on the horizontal surface ?

SOLUTION.— $4 \times 6 \times 26 \times .03617 = 22.57$ pounds, the upward pre sure due to the weight of the water.

$6 \times 4 \times 16 = 384$ pounds, the upward pressure due to the outsid pressure of 16 pounds per square inch.

The total upward pressure $= 384 + 22.57 = 406.57$ pounds. Ans.

2176. Lateral (Sideways) Pressure.—Suppose th top of the vessel shown in Fig. 733 is 10 inches square

FIG. 733.

and that the projections at a and b ar 1 inch × 1 inch, and 10 inches long.
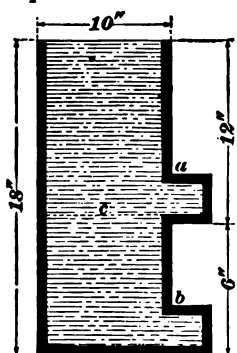
The pressure per square inch on th bottom of the vessel, due to the weigh of the liquid, is $1 \times 1 \times 18 \times$ the weigh of a cubic inch of the liquid.

The pressure at a depth equal to th distance of the upper surface b is $1 \times 1 \times 17 \times$ the weight of a cubic inch of th liquid.

Since both of these pressures are trans mitted in every direction, they are als transmitted laterally (sideways), and the *pressure per uni of area on the projection b is a mean between the two*, an equals $1 \times 1 \times 17\frac{1}{2} \times$ the weight of a cubic inch of the liquid

To find the lateral pressure on the projection a, imagin that the dotted line c is the bottom of the vessel; then th conditions would be the same as in the preceding case, ex cept that the depth is not so great.

The lateral pressure on a is thus seen to be $1 \times 1 \times 11\frac{1}{2} \times$ the weight of a cubic inch of the liquid.

2177. General law for lateral pressure :

The pressure upon any vertical surface, due to the weight oj

the liquid, is equal to the weight of a prism of the liquid whose base has the same area as the vertical surface, and whose altitude is the depth of the center of gravity of the vertical surface below the level of the liquid.

Any additional pressure is to be added, as in the previous cases.

EXAMPLE.—A well 3 feet in diameter and 20 feet deep is filled with water; what is the pressure on a strip of the wall 1 inch wide, the center of which is 1 foot from the bottom? What is the pressure on the bottom? What is the upward pressure per square inch, 2 feet 6 inches from the bottom?

SOLUTION.— $1 \times 36 \times 3.1416 = 113.1$ sq. in., the area of the strip. $113.1 \times 19 \times 12 \times .03617 = 932.71$ pounds, total pressure upon the strip. Ans.

The pressure per square inch would be $\dfrac{932.71}{113.1} = 8.247$ pounds, nearly.

$36 \times 36 \times .7854 \times 20 \times 12 \times .03617 = 8,836$ pounds, the pressure on the bottom. Ans.

$20 - 2.5 = 17.5$. $1 \times 17.5 \times 12 \times .03617 = 7.596$ pounds, the upward pressure per square inch, 2 feet 6 inches from the bottom. Ans.

2178. The effects of lateral pressure are illustrated in Fig. 734; *e* is a tall vessel having a stop-cock near its base, and arranged to float upon the water, as shown. When this
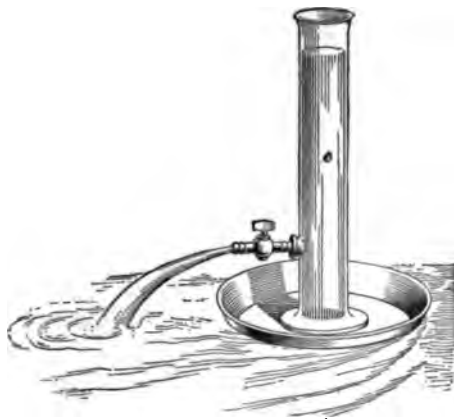
FIG. 734.

vessel is filled with water, the lateral pressures at any two points of the surface of the vessel and opposite to each other are equal. Being equal, and acting in opposite directions,

they destroy each other, and no motion can result; but if the stop-cock be opened, there will be no resistance to that pressure acting on the surface equal to the area of the opening, and it will cause the water to flow out, while its equal and opposite force will cause the vessel to move backwards through the water in a direction opposite to that of the spouting water.

2179. The laws of liquid pressure given in the preceding articles may be embraced in the following formula :

$$P = a\,(d\,w + p),\qquad \textbf{(164.)}$$

where a = area of a submerged surface in square inches;

d = distance in inches of center of gravity of surface from surface of liquid;

w = weight of a cubic inch of the fluid in pounds ;

p = pressure on surface of liquid in pounds per square inch;

P = total pressure on submerged surface in pounds.

2180. Since the pressure on the bottom of a vessel due to the weight of the liquid is dependent only upon the height of the liquid, and not upon the shape of the vessel, it follows that if a vessel has a number of radiating tubes, as shown in Fig. 735, the water in each tube will be on the same level, no matter what may be the shape of the tubes.
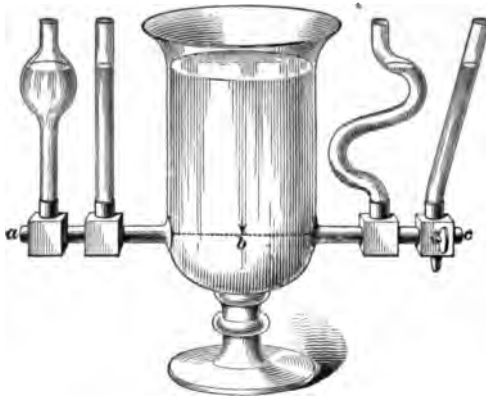
FIG. 735.

For, if the water were higher in one tube than in the others, the downward pressure on the bottom due to the height of the water in this tube would be greater than that due to the height of the water in the other tubes. Consequently, the upward pressure would also be greater, the equilibrium would be destroyed, and the water would flow from this tube into the

vessel, and rise in the other tubes until it was at the same level in all, when it would be in equilibrium. This principle is expressed in the familiar saying, *water seeks its level.*

This explains why city water-reservoirs are located on high elevations, and why water on leaving the hose-nozzle spouts so high.

If there were no resistance by friction and air, the water would spout to a height equal to the level of the water in the reservoirs. If a long pipe whose length was equal to the vertical distance between the nozzle and the level of the water in the reservoir were attached to the nozzle, the water would just reach the end of the pipe. If the pipe were lowered slightly, the water would trickle out. Fountains, canal-locks, and artesian wells are examples of the application of this principle.

EXAMPLE.—The water-level in a city reservoir is 150 feet above the level of the street; what is the pressure of the water per square inch on the hydrant?

SOLUTION.— $1 \times 150 \times 12 \times .03617 = 65.106$ pounds per square inch. Ans.

2181. In Fig. 736, let the area of the piston *a* be 1 square inch, of *b* 40 square inches. According to Pascal's law, 1 pound placed upon *a* will balance 40 pounds placed upon *b*.

Suppose that *a* moves downwards 10 inches, then 10 cubic inches of water will be forced into the tube *b*. This will be distributed over the entire area of the tube *b*, in the form of a cylinder, whose cubical contents must be 10 cubic inches, whose base has an area of 40 square inches, and whose altitude must be

FIG. 736.

$\frac{10}{40} = \frac{1}{4}$ of an inch; that is, a movement of 10 inches of the piston *a* will cause a movement of $\frac{1}{4}$ of an inch in the piston *b*.

This is analogous to the old principle of machines: *The power, multiplied by the distance through which it moves*
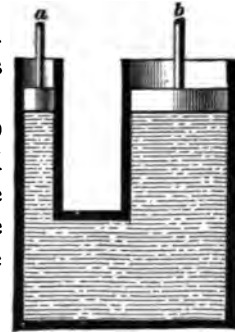
equals the weight multiplied by the distance through which it moves.

2182. The foregoing principles are made use of in the hydraulic press represented in Fig. 737. As the lever *O* is
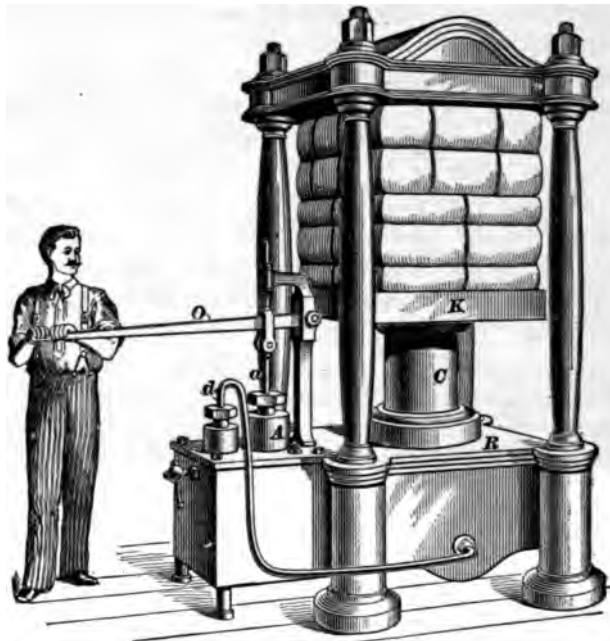
FIG. 737.

depressed, the piston *a* is forced down upon the water in the cylinder *A*. The water is forced through the bent tube *d* into the cylinder in which the large piston *C* works, and causes *C* to rise, thus lifting the platform *K*, and compressing the bales. If the area of *a* be ½ sq. in., and that of *C* be 50 sq. in., it is evident, from the explanation of Fig. 736, that a force of 10 pounds on the piston *a* will lift a load of $\frac{10}{.5} \times 50 = 1{,}000$ pounds on the piston *C*. If, now, the length of the lever between the hand and the fulcrum is 10 times the length between the fulcrum and the piston *a*, a force of 10 pounds on the end of the lever will exert **100 pounds on *a*,** and, therefore, **10,000 pounds on *C*.**

Applications of this principle are seen in the hydraulic machines used for forcing locomotive drivers on their axles, etc., and for testing the strength of boiler-shells.

EXAMPLE.—A suspended vertical cylinder is tested for the tightness of its heads by filling it with water. A pipe whose inside diameter is ¼ of an inch, and whose length is 20 feet, is screwed into a hole in the upper head, and then filled with water; what is the pressure per square inch on each head, if the cylinder is 40 inches in diameter and 60 inches long ?

SOLUTION.—Area of heads $= 40^2 \times .7854 = 1,256.64$ sq. in.

The pressure per square inch on the bottom head due to the weight of the water in the cylinder $= 1 \times 60 \times .03617 = 2.17$ pounds. $(\frac{1}{4})^2 \times .7854 = .04909$ sq. in., the area of the pipe.

$.04909 \times 20 \times 12 \times .03617 = .426$ pound $=$ the weight of water in pipe $=$ the pressure on a surface area of .04909 sq. in.

The pressure per square inch due to the water in the pipe is $\frac{1}{.04909} \times .426 = 8.68$ pounds per square inch upon the upper head. Ans.

The pressure per square inch on the lower head is $8.68 + 2.17 = 10.85$ pounds. Ans.

EXAMPLE.—In the last example, if the pipe be fitted with a piston weighing ¼ of a pound, and a 5-pound weight be laid upon it, what will be the pressure upon the upper head ?

SOLUTION.—In addition to the pressure of .426 pound on the area of .04909 sq. in., there is now an additional pressure upon this area of $5 + \frac{1}{4} = 5.25$ pounds, and the total pressure upon this area is $.426 + 5.25 = 5.676$ pounds. Ans.

The pressure per square inch is $\frac{1}{.04909} \times 5.676 = 115.6$ pounds.

BUOYANT EFFECTS OF WATER.

2183. In Fig. 738 is shown a 6-inch cube, entirely submerged in water. The lateral pressures are equal and in opposite directions. The upward pressure $= 6 \times 6 \times 21 \times .03617$; the downward pressure $= 6 \times 6 \times 15 \times .03617$, and the difference $= 6 \times 6 \times 6 \times .03617 =$ the volume of the cube in cubic inches $\times$ the weight of 1 cubic inch of water. That is, the upward pressure exceeds the downward pressure by the weight of a volume of water equal to the volume of the body.
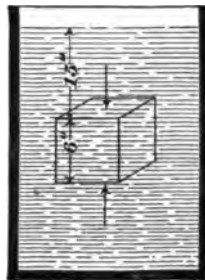
FIG. 738.

2184. This excess of upward pressure over the down—ward pressure acts against gravity; consequently, *if a body be immersed in a fluid, it will lose in weight an amount equal to the weight of the fluid it displaces.* This is called the **prin—ciple of Archimedes,** because it was first stated by him.

This principle may be experimentally demonstrated with the beam-scales, as shown in Fig. 739.

From one scale-pan suspend a hollow cylinder of metal *t* and below that a solid cylinder *a*, of the same size as the hollow part·of the upper cylinder. Put weights in the other scale-pan until they exactly balance the two cylinders. If *a* be immersed in water, the scale-pan containing the weights will descend, showing that *a* has lost some of its weight. Now fill *t* with water, and the volume of water that can be poured into *t* will equal that displaced by *a*. The scale-pan that con—tains the weights will gradually rise until *t* is filled, when the scales balance again.
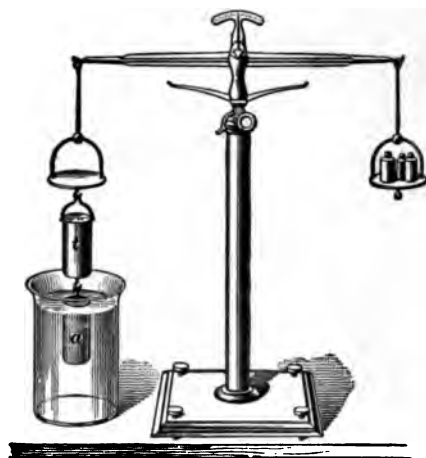
FIG. 739.

If the immersed body be lighter than the liquid, the up—ward pressure will cause it to rise and extend partly out of the liquid, until the weight of the body and the weight of the liquid displaced are equal. If the immersed body be heavier than the liquid, the downward pressure, plus the weight of the body, will be greater than the upward pressure, and the body will fall downwards, until it touches bottom or meets an obstruction. If the weights of equal volumes of the liquid and the body are equal, the body will remain stationary, and be in equilibrium in any position or depth beneath the surface of the liquid.

An interesting experiment in confirmation of the above

facts may be performed as follows: Drop an egg into a glass jar filled with fresh water. The mean density of the egg being a little greater than that of water, it will fall to the bottom of the jar. Now dissolve salt in the water, stirring it so as to mix the fresh and salt water. The salt water will presently become denser than the egg, and the egg will rise. Now, if fresh water be poured in until the egg and water have the same density, the egg will remain stationary in any position that it may be placed below the surface of the water.

HYDROKINETICS.

FLOW OF WATER THROUGH SHORT TUBES.

2185. Hydrokinetics, also called **hydrodynamics and hydraulics,** treats of water in motion. The velocity is not the same at all points of the flow, unless all cross-sections of the pipe or canal are equal. That *velocity* which, being *multiplied by the area of the cross-section of the stream,* will equal the total quantity *discharged* is called the **mean velocity.**

Let Q = the quantity which passes any section in one second;

A = the area of the section;

v = the mean velocity in feet per second.

Then, $$Q = A v, \qquad (165.)$$

and $$v = \frac{Q}{A}. \qquad (166.)$$

EXAMPLE.—The area of a certain cross-section of a stream is 27.9 square inches; the velocity of the water through this section is 51 feet per second; what is the quantity discharged in cubic feet ?

SOLUTION.—Applying formula **165,** $Q = \dfrac{27.9}{144} \times 51 = 9.9$ cubic feet per second. Ans.

EXAMPLE.—In the last example, what would the velocity have been to discharge the same quantity had the area of the cross-section been 16 square inches ?

SOLUTION.—Applying formula **166,** $V = \dfrac{9.9}{\frac{36}{144}} = \dfrac{9.9 \times 144}{36} = 39.6$ ft. per sec. Ans.

2186. Velocity of Efflux.—If a small aperture be made in a vessel containing water, the velocity with which the water issues from the vessel is the same as if it had fallen from the level of the surface to the level of the aperture, all resistances being neglected. This velocity is called the **velocity of efflux.**
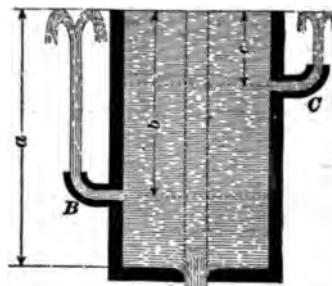
The vertical height of the level surface of the water above the center of the aperture is called the **head.** In Fig. 740, a is the head for the aperture A; b is the head for the aperture B ; and c is the head for the aperture C.

FIG. 740.

Let $v =$ the velocity of efflux in feet per second;

$h =$ the head in feet at the aperture considered.

Then, the theoretical velocity of efflux is expressed by the formula

$$v = \sqrt{2gh}. \qquad \textbf{(167.)}$$

Here $g = 32.16$; that is, *the velocity of efflux is the same as if the same weight of water had fallen through a height equal to its head.*

Were it not for the resistance of the air, friction, and the effect of the falling particles, the issuing water would spout to the level of the water in the vessel; that is, to a height equal to its head.

EXAMPLE.—A small orifice is made in a pipe 50 feet below the water-level; what is the velocity of the issuing water?

SOLUTION.—Applying formula **167,** $v = \sqrt{2 \times 32.16 \times 50} = 56.7$ feet per second. Ans.

From the above formula, as in the laws of falling bodies,

$$h = \frac{v^2}{2g}. \qquad \textbf{(168.)}$$

Here, h is called the *head due to the velocity v.* Consequently, if the velocity of efflux is known, the head can be found.

EXAMPLE.—An issuing jet of water has a velocity of 60 feet per second; what must be the head to give it this velocity?

SOLUTION.—Applying formula **168**, $h = \dfrac{60^2}{2 \times 32.16} = 55.97$ feet. Ans.

2187. Suppose that a tall vessel is fitted with a piston, and has an orifice near the bottom fitted with a stop-cock. If an additional pressure be applied to the piston, it is evident that the velocity of efflux will be increased.

Let p be the pressure per unit of area at the level of the water, due to the additional pressure on the piston. If the unit of area is one square inch, the height of a column of water that would cause a pressure equal to p would be $\dfrac{p}{.434}$ feet.

If the unit of area is in square feet, the height of a column of water would be $\dfrac{p}{62.5}$ feet. Denote this height corresponding to the additional pressure by h_1. The original head of the water in the vessel is h; hence, $h_1 + h =$ the total head, and the velocity of efflux, when the cock is opened, will be

$$v = \sqrt{2g(h_1 + h)}. \qquad \textbf{(169.)}$$

The total head $h_1 + h$ is called the **equivalent head,** and must, in all cases, be reduced to feet before substituting in the formula.

EXAMPLE.—The area of a piston fitting a vessel filled with water is 27.36 square inches. The total pressure on the piston is 80 pounds; the weight of the piston is 25 pounds, and the head of the water at the level of the orifice is 6 feet 10 inches; what is the velocity of efflux, assuming that there are no resistances?

SOLUTION.— $80 + 25 = 105$ pounds $=$ the total pressure on the upper surface of the liquid. $\dfrac{105}{27.36} = 3.838$ pounds per square inch.

$\dfrac{3.838}{.03617} = 106.11 =$ head in inches due to the pressure of 105 pounds.

$\dfrac{106.11}{12} = 8.84$ ft. $= h_1$.   6 feet 10 inches $= 6.8333$ feet $= h$.

Hence, applying formula **169,**

$$v = \sqrt{2g\,(8.84 \div 6.8333)} = \sqrt{2 \times 32.16 \times 15.6733} = 31.75 \text{ feet per second.}$$
$$\text{Ans.}$$

2188. When water issues from the side of a vessel, it is subjected to the same laws that govern projectiles. The range may be calculated in the same manner by taking the *velocity of efflux* as the *initial velocity* of the projectile.

The range may be calculated more conveniently by the following formula:

$$R = \sqrt{4\,h\,y}, \qquad (170.)$$

in which R is the range, h is the head or equivalent head at the level of the orifice, and y is the vertical height of the orifice above the point where the water strikes. In Fig. 741, the upper surface of the water is free. For the orifice E, $h = B\,E$, and $y = E\,A$; for the orifice C, $h = B\,C$, and $y = C\,A$.

The greatest range is obtained when $h = y$; that is, when the orifice is half way between the upper surface of the water and the level of the place where the stream strikes.
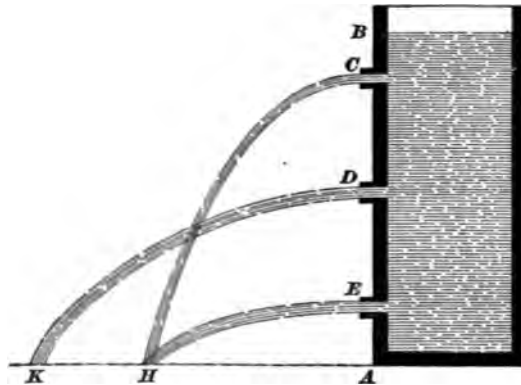
FIG. 741.

If two orifices are situated equally distant from the middle orifice, giving the greatest range, as C and E in Fig. 741, the ranges of water issuing from them will be equal.

EXAMPLE.—The vertical height above the ground of the surface of the water in a vessel is 12 feet. If an orifice is situated 4 feet from

the upper surface, what is the range? Where is the other point of equal range? What is the greatest range?

SOLUTION.—Applying formula **170**, $R = \sqrt{4 \times 4\,(12-4)} = 11.31$ feet, nearly; greatest range $= \sqrt{4 \times 6 \times 6} = 12$ feet. $6-4=2$; hence, the point of equal range is $6+2=8$ feet below the surface of the water. Ans.

PROOF.—Range $= \sqrt{4\,hy} = \sqrt{4 \times 8 \times 4} = 11.31$ feet as before.

2189. When the water flows through an orifice in the bottom of the vessel, of large size compared with the area of the base, a different rule must be used from that given above. In Fig. 742, suppose that the area of the orifice in the bottom of the vessel is a, and that the area of the bottom is A; then the velocity v is expressed by the formula

$$v = \sqrt{\dfrac{2\,g\,h}{1-\dfrac{a^2}{A^2}}}. \qquad (\mathbf{171.})$$
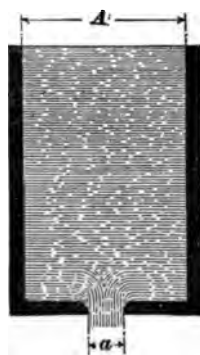
FIG. 742.

That is, *the velocity of efflux from the bottom of a vessel in feet per second equals the square root of 2 g times the head, divided by 1, minus the ratio of the square of the area of the orifice, to the square of the area of the bottom.*

If the area of the orifice is not more than $\frac{1}{20}$ of the area of the cross-section of the vessel, use formula **167.** That is, *the velocity of efflux from a small orifice, not larger than $\frac{1}{20}$ of the cross-sectional area of the vessel, equals the square root of 2 g times the head.*

EXAMPLE.—A vessel has a rectangular cross-section of 11×14 inches; the upper surface of the water is 14 feet above the bottom. If an orifice, 4 inches square, is made in the bottom of the vessel, what will be the velocity of efflux?

SOLUTION.—Area of the cross-section is $14 \times 11 = 154$ sq. in. Area of orifice is $4 \times 4 = 16$ sq. in., $\frac{16}{154} = \frac{1}{9.625}$. Since the area of the orifice is greater than $\frac{1}{20}$ of the area of the bottom, apply formula **171.**

$$v = \sqrt{\dfrac{2\,g\,h}{1-\dfrac{a^2}{A^2}}} = \sqrt{\dfrac{2 \times 32.16 \times 14}{1-\dfrac{16^2}{154^2}}} = 30.17 \text{ feet per second. Ans.}$$

EXAMPLE.—If the orifice had been 2 inches square in the above
example, what would have been the velocity of efflux? Also, if it had
been 8 inches square?

SOLUTION.— $2 \times 2 = 4$ sq. in., or the area of the orifice. $\frac{4}{154} =$
$\frac{1}{38.5}$. Since the area of the orifice is less than $\frac{1}{10}$ of the area of the
vessel, apply formula **167,**

$$v = \sqrt{2gh} = \sqrt{2 \times 32.16 \times 14} = 30.008 \text{ feet per second. Ans.}$$

$8 \times 8 = 64$ sq. in., or the area of the orifice in the second case; then
applying formula **171,**

$$v = \sqrt{\frac{2gh}{1 - \frac{a^2}{A^2}}} = \sqrt{\frac{2 \times 32.16 \times 14}{1 - \frac{64^2}{154^2}}} = 32.99 \text{ feet per second; prac-}$$

cally, 33 feet per second. Ans.

2190. The Contracted Vein.—When water issues
from an orifice in a thin plate (see
Fig. 743), or from a square-edged ori-
fice (see Fig. 744), the stream is con-
tracted a short distance from the
orifice, and expands again to the full
size of the orifice. The point *at*
which the contraction is greatest *is*
at a distance from the orifice equal *to*
the diameter of the orifice. In con-
sequence of this contraction, the
velocity of efflux is slightly reduced
from the theoretical value, and the quantity discharged is
greatly reduced. This contrac-
tion is called the **contracted
vein,** a name given to it by
Sir Isaac Newton.
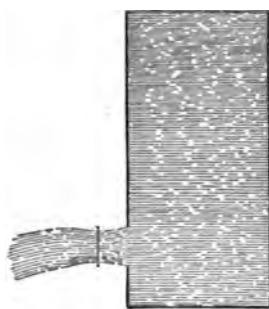
FIG. 743.

For ordinary purposes, the
actual velocity of efflux may be
taken as 98% of the theoretical
values calculated by the prece-
ding rules.

The actual velocity of efflux
from a small orifice is expressed by the formula

FIG. 744.

$$v = .98\sqrt{2gh}. \qquad (172.)$$
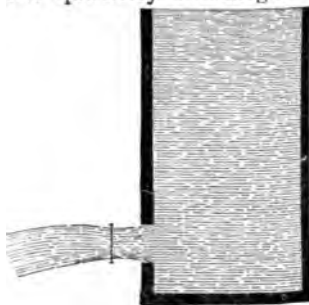
EXAMPLE.—What is the actual velocity of discharge from a small, square-edged orifice in the side of a vessel, if the head is 20 feet ?

SOLUTION.—Applying formula 172,

$v = .98 \sqrt{2\,g\,h} = .98 \sqrt{2 \times 32.16 \times 20} = 35.15$ feet per second.   Ans.

2191. The diameter of the contracted vein at its smallest section is about .8 of the diameter of the orifice, and its area is about $.8 \times .8 = .64$ of the area of the orifice. In Art. **2185,** it was stated that the quantity discharged in cubic feet per- second was equal to the area of the section multiplied by the mean velocity, or $Q = A\ v$. This was the theoretical value; *the actual value is the area of the contracted vein multiplied by the actual velocity of efflux*, or $Q = .64\ A \times .98\ v = .627\ A\ v$; that is, the actual discharge is about .627 of the theoretical discharge.

This number, .627, is called the **coefficient of efflux.**

The coefficient of efflux varies somewhat according to the head, and the size and shape of the orifice; but for square-edged orifices, or for orifices in thin plates, its average value may be taken as .615. Hence,

Rule.—*The actual quantity discharged is .615 times the theoretical amount,*

or, $Q = .615\ A\ v.$ **(173.)**

EXAMPLE.—The theoretical discharge from a certain vessel is 12.4 cubic feet per minute; what is the amount actually discharged per second ?

SOLUTION.— $12.4 \times .615 = 7.626$ cubic feet per minute; $\dfrac{7.626}{60} =$ .1271 cubic foot per second.   Ans.

2192. If the water discharges through a short tube whose length is from $1\frac{1}{2}$ to 3 times the diameter of the orifice (see Fig. 745), the discharge is increased. From a large number of experiments made by different persons, the coefficient of efflux for a short tube may

FIG. 745.

be taken as .815; that is, the actual discharge may be taken
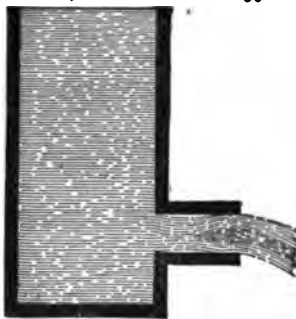
as .815 times the theoretical discharge through an orifice of the same size. If the inside edges of the tube are well round
and the tube is conical, as shown in Fig. 746, there will be no contraction, and the coefficient of discharge may be taken as .97; that is, the actual discharge through a tube of this form is .97 times the theoretical discharge through an orifice whose area is the same as the area of the end of the tube.
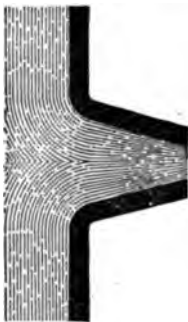
2193. If in a compound mouthpiece or tube, such as is shown in Fig. 747, a b, the narrowest part, be taken as the diameter of the orifice, the coefficient of discharge may be taken as 1.5526; that is, the actual discharge through a compound mouthpiece of this shape is 1.5526 times the theoretical discharge through an orifice whose area is the same as the area of the smallest section of the mouthpiece.

FIG. 746.

2194. When the upper surface of the water remains at the same height above the orifice, there is said to be a *constant head.* The velocity of efflux varies for different points in

FIG. 747.

the orifice; it is greater at the bottom of the orifice than at the top, since the head is greater at the bottom than at the top. A mean velocity may be obtained by *dividing the quantity of water discharged in cubic feet per second by the area of the orifice;*
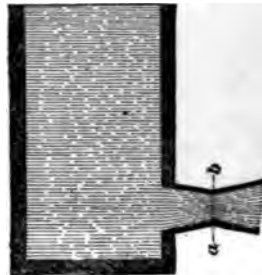
or, $$v_m = \frac{Q}{A}.$$ (See formula **166.**)

2195. Let

Q = theoretical number of cubic feet discharged per second;

v_m = mean velocity through orifice;

A = area of orifice;

h = theoretical head necessary to give a mean velocity v_m;

Q_a = actual quantity discharged in cubic feet per second.

Then, for an orifice in a thin plate, or a square-edged orifice (the hole itself may be of any shape—triangular, square, circular, etc.—but the edges must not be rounded), the actual quantity discharged is

$$Q_a = .615 \ Q, \text{ or}$$
$$= .615 \ A \ v_m, \text{ or}$$
$$= .615 \ A \ \sqrt{2 \ g \ h}. \qquad \textbf{(174.)}$$

That is, *the actual quantity discharged through a square-edged orifice, or through a thin plate, is .615 times the theoretical discharge, and equals .615 multiplied by the area of the orifice, multiplied by the mean velocity; or equals .615 multiplied by the area of the orifice, multiplied by the square root of 2 g times the theoretical head, corresponding to the theoretical mean velocity.*

For a discharge through a short tube, as shown in Fig. 745,

$$Q_a = .815 \ Q = .815 \ A \ v_m = .815 \ A \ \sqrt{2 g h}. \qquad \textbf{(175.)}$$

For a discharge through a mouthpiece, as shown in Fig. 746,

$$Q_a = .97 \ Q = .97 \ A \ v_m = .97 \ A \ \sqrt{2 \ g \ h}. \qquad \textbf{(176.)}$$

For a discharge through the compound mouthpiece, as shown in Fig. 747, the area of the orifice being taken as the area of the smallest section,

$$Q_a = 1.5526 \ Q = 1.5526 \ A \ v_m = 1.5526 \ A \ \sqrt{2 \ g \ h}. \qquad \textbf{(177.)}$$

In these four formulas it is assumed that the head remains constant.

WEIRS.

2196. The **weir** is a device universally used for measuring the discharge of water. It is a rectangular orifice through which the water flows.

2197. In Fig. 748 is represented a weir in which th top of the weir (orifice) is level with the upper surfac of the water flowing throug it. By means of highe mathematics, it has bee found that the *theoretica mean velocity* v_m is equal $\frac{2}{3} \sqrt{2 g h}$.

FIG. 748.

2198. If $d =$ the dept of the opening in feet, and its breadth in feet, the area the opening $A = d\,b$, and the theoretical discharge is $Q =$ $d\,b\,v_m = \frac{2}{3}\,d\,b\,\sqrt{2\,g\,d}$, the head for this case being taken as

The actual discharge is

$$Q_a = .615\,Q = .615\,d\,b \times \tfrac{2}{3}\sqrt{2\,g\,d} = .41\,b\sqrt{2\,g\,d^3}. \qquad (178$$

That is, *the actual discharge through a weir in cubic fe per second, whose top is on a level with the upper surface the water, is equal to .41 multiplied by the breadth of th weir, multiplied by the square root of 2 g times the cube of th depth of the weir.* All dimensions are to be taken in feet.

EXAMPLE.—A weir like the one represented in Fig. 748 has a dept $d = 18$ inches, and a breadth $b = 30$ inches; what is the actual dischar per minute in cubic feet?

SOLUTION.—Applying formula **178**,

$$Q_a = .41\,b\sqrt{2\,g\,d^3} = .41 \times \tfrac{30}{12} \times \sqrt{2 \times 32.16 \times (\tfrac{18}{12})^3} = 15.1 \text{ cubic feet per secon}$$
$15.1 \times 60 = 906$ cubic feet per minute. Ans.

2199. To obtain the mean velocity v'_m, divide the actu discharge by the area of the weir.

Thus, $\qquad v'_m = \dfrac{Q_a}{A} = \dfrac{Q_a}{b\,d}.$ $\qquad$ **(179.)**

EXAMPLE.—What is the mean velocity in feet per second of th water in the last example?

SOLUTION.—Applying formula **179**,

$$v_m = \frac{Q_a}{b\,d} = \frac{15.1}{2\frac{1}{2} \times 1\frac{1}{2}} = \frac{15.1}{3.75} = 4.027 \text{ feet per second.} \quad \textbf{Ans.}$$

2200. It should be kept in mind that a weir is but a rectangular opening. It is a special name given to a rectangular orifice. Some writers use the term **rectangular notch** instead of weir.
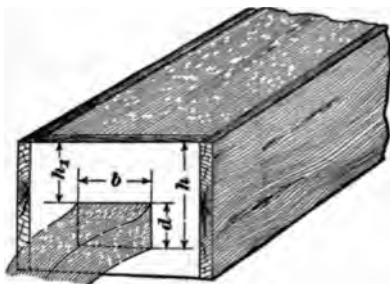
2201. In Fig. 749 is represented a weir whose top is below the level of the upper surface of the water. If h_1 is the depth in feet of the top of the weir below the surface of the water, and h is the depth in feet of the bottom of the weir below the surface of the water, the actual discharge Q_a in cubic feet per second is

$$Q_a = .41\, b\sqrt{2g}\left[\sqrt{h^3} - \sqrt{h_1^3}\right]. \qquad \textbf{(180.)}$$

That is, *the actual discharge through a weir, whose top is h_1 feet, and whose bottom is h feet, below the surface of the water, is equal to .41 times the breadth of the weir multiplied by the square root of 2 g times the difference of the square roots of the cubes of the depth of the bottom of the weir, and the depth of the top of the weir.*

EXAMPLE.—A weir like that shown in Fig. 749 has a depth $d = 2$ feet, and a breadth $b = 3$ feet. The depth of the top below the surface of the water is 5 feet; what is the discharge in cubic feet per minute?

SOLUTION.— $h_1 = 5.$ $h = 5 + 2 = 7.$ Hence, applying formula **180,**
$$Q_a = .41\, b\sqrt{2g}\left[\sqrt{h^3} - \sqrt{h_1^3}\right] =$$
$.41 \times 3 \times \sqrt{2 \times 32.16} \times \left[\sqrt{7^3} - \sqrt{5^3}\right] = 72.41$ cubic feet per second $=$
$72.41 \times 60 = 4,344.6$ cubic feet per minute. Ans.

EXAMPLE.—What is the mean velocity in the last example?

SOLUTION.—Applying formula **179,**
$$v_m = \frac{Q_a}{b\,d} = \frac{72.41}{2 \times 3} = 12.07 \text{ feet per second. Ans.}$$

FLOW OF WATER IN PIPES.

LOSS DUE TO FRICTION.

2202. When water flows from one reservoir to another through a pipe, the velocity of efflux is considerably less than the theoretical velocity due to the head. This loss is

due to several causes, but is principally caused by the f**——i**-
tion of the water against the inside surface of the pi **pe**
This friction varies directly as the length of the pipes, and **in**-
versely as the diameter; that is, the friction in a pipe **200**
feet long is twice as much as in a pipe 100 feet long, **and**
the friction in a pipe 4 inches in diameter is only half   **as**
much as in a pipe 2 inches in diameter, the velocity rema **in**-
ing the same in both cases. *The friction also varies* **with**
the velocity.

MEAN VELOCITY OF DISCHARGE.

2203.   The following formulas apply to straight cyl **in**-
drical pipes of uniform diameter:

Let v_m = mean velocity of discharge in feet per secon**d ;**
h = total head in feet = the vertical distance **be**-
tween the level of the water in the reserv**oir**
and the point of discharge;
l = length of pipe in feet;
d = diameter of pipe in inches;
f = coefficient of friction.

Then, for straight cylindrical pipes of uniform diamet**er,**
the mean velocity of efflux may be calculated by **the**
formula

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l + .125\,d}}. \qquad \textbf{(181.)}$$

That is, *the mean velocity of discharge equals 2.315 times*
the square root of the head in feet, multiplied by the dia—*m*-
eter in inches, divided by the coefficient of friction, multi-
plied by the length in feet, plus .125 of the diameter of the
pipe in inches.

The head is always taken as the vertical distance betwe**en**
the point of discharge and the level of the water at t**he**
source, or point from which it is taken, and is always mea**s**-
ured in feet.   It matters not how long the pipe is, wheth**er**
vertical or inclined, whether straight or curved, n**or**
whether any part of the pipe goes below the level of the
point of discharge, the head is always measured as stated
above.

EXAMPLE.—What is the mean velocity of efflux from a 6-inch pipe 5,780 feet long, if the head is 170 feet ?   Take $f = .021$.

SOLUTION.—Applying formula **181**,

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l + .125\,d}} = 2.315 \sqrt{\frac{170 \times 6}{.021 \times 5,780 + (.125 \times 6)}} = 6.69$$

feet per second.   Ans.

2204.   When the pipe is very long, compared with the diameter, as in the above example, the following formula may be used:

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l}}, \qquad \textbf{(182.)}$$

in which the letters have the same meaning as in the preceding formula.   This formula may be used when the length of the pipe exceeds 10,000 times its diameter.

EXAMPLE.—In the preceding example, calculate the value of v_m by using formula **182.**

SOLUTION.—

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l}} = 2.315 \sqrt{\frac{170 \times 6}{.021 \times 5,780}} = 6.71 \text{ feet per second.   Ans.}$$

THE ACTUAL HEAD.

2205.   The **actual head** necessary to produce a certain velocity v_m may be calculated by the formula

$$h = \frac{f\,l\,v_m^2}{5.36\,d} + .0233\,v_m^2. \qquad \textbf{(183.)}$$

That is, *the total head in feet necessary to produce a velocity of efflux v_m in a straight cylindrical pipe is equal to the coefficient of friction multiplied by the length of the pipe in feet, multiplied by the square of the mean velocity of efflux in feet per second, divided by 5.36 times the diameter of the pipe in inches, plus .0233 times the square of the mean velocity.*

EXAMPLE.—A 7-inch pipe 6,000 feet long is required to deliver water with a velocity of 7 feet per second; what must be the necessary head ?   Assume $f = .026$.

SOLUTION.—Applying formula **183**,

$$h = \frac{f \, l \, v_m^2}{5.36 \, d} - .0233 \, v_m^2 = \frac{.036 \times 6,000 \times 49}{5.36 \times 7} + .0233 \times 49 = 204.87 \text{ feet}$$
nearly. Ans.

THE QUANTITY DISCHARGED FROM PIPES.

2206. The formulas just given are made use of in ascertaining the quantity of water that will be discharged from a pipe in a given time with a given head. This readily found by substituting the value of v_m for v in formula **165**; thus, $Q = A \, v_m$.

Since $A = .7854 \, d^2$ = area of the inside of the pipe, the quantity discharged can be readily calculated as soon as v is known. This method gives the discharge in cubic feet per second, when the diameter d is taken in feet.

One cubic foot contains 7.48 gallons; hence, when d taken in feet,

$$Q = .7854 \, d^2 \, v_m \times 7.48 \text{ gallons.}$$

If d is taken in inches,

$$Q = \frac{.7854 \, d^2}{144} \, v_m \times 7.48;$$

or, $\qquad Q = .0408 \, d^2 \, v_m.$ **(184.)**

That is, *the discharge in gallons per second equals .040 times the square of the diameter of the pipe in inches, multiplied by the mean velocity of efflux in feet per second.*

EXAMPLE.—What is the discharge in gallons per minute from 6-inch pipe, if the mean velocity of efflux is 5.6 feet per second?

SOLUTION.—Applying formula **184**,

$$Q = .0408 \, d^2 \, v_m = .0408 \times 36 \times 5.6 = 8.225 \text{ gallons per second.}$$
$$8.225 \times 60 = 493.5 \text{ gallons per minute. Ans.}$$

2207. If the diameter of the pipe and the discharge are known, the mean velocity can be readily found by the formula

$$v_m = \frac{24.51 \, Q}{d^2}. \qquad \textbf{(185.)}$$

That is, *the mean velocity of discharge equals 24.51 times the number of gallons discharged per second, divided by the square of the diameter of the pipe in inches.*

EXAMPLE.—A 5-inch pipe is discharging 360 gallons per minute; what is the mean velocity of efflux ?

SOLUTION.— $\frac{360}{60} = 6$ gallons discharged per second. Hence, applying formula **185,**

$$v_m = \frac{24.51 \times Q}{d^2} = \frac{24.51 \times 6}{25} = 5.882 \text{ feet per second. Ans.}$$

2208. If the head, the length of the pipe, and the diameter of the pipe are given, to find the discharge, use the formula

$$Q = .09445\, d^2 \sqrt{\frac{h\,d}{f\,l + .125\,d}}. \qquad \textbf{(186.)}$$

That is, *the discharge in gallons per second equals .09445 times the square of the diameter of the pipe in inches, multiplied by the square root of the head in feet times the diameter of the pipe in inches, divided by the coefficient of friction times the length of the pipe in feet, plus .125 times the diameter of the pipe in inches.*

2209. To find the value of f, calculate v_m by formula **182,** assuming that $f = .025$, and get the final value of f from the following table:

TABLE 45.

$v_m =$	0.1	0.2	0.3	0.4	0.5	0.6
$f =$	.0686	.0527	.0457	.0415	.0387	.0365

$v_m =$	0.7	0.8	0.9	1	$1\frac{1}{4}$	$1\frac{1}{2}$
$f =$	.0349	.0336	.0325	.0315	.0297	.0284

$v_m =$	2	3	4	6	8	12
$f =$	.0265	.0243	.023	.0214	.0205	.0193

EXAMPLE.—The length of a pipe is 6,270 feet, its diameter is 8 inches, and the total head at the point of discharge is 215 feet; how many gallons are discharged per minute ?

SOLUTION.—Using formula **182**,

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l}} = 2.315 \sqrt{\frac{215 \times 8}{.025 \times 6,270}} = 7.67 \text{ feet per second, nearly}$$

For $v_m = 6$, $f = .0214$, and for $v_m = 8$, $f = .0205$.

$.0214 - .0205 = .0009 =$ difference for a difference in the v_m's of 2 ft per sec.

$.0009 \div 2 = .00045 =$ difference for a difference in the v_m's of 1 ft per sec.

$7.67 - 6 = 1.67$. $.00045 \times 1.67 = .0007515 =$ amount to be subtracted from .0214 to obtain f for $v_m = 7.67$.   Using but five decimal places, $.0214 - .00075 = .02065 = f$ for $v_m = 7.67$.

Hence, applying formula **186**,

$$Q = .09445\, d^2 \sqrt{\frac{h\,d}{f\,l + .125\,d}} = .09445 \times 8^2 \sqrt{\frac{215 \times 8}{.02065 \times 6,270 + .125 \times 8}} =$$

21.95 gallons per second, nearly.

$21.95 \times 60 = 1,317$ gallons per minute.   **Ans.**

2210.   If it is desired to find the head necessary to give a discharge of a certain number of gallons per second through a pipe whose length and diameter are known, calculate the mean velocity of efflux by using formula **185** Find the value of f from Table 45, corresponding to this value of v_m; substitute these values of f and v_m in formula **183**, and calculate the head.

EXAMPLE.—A 4-inch pipe, 2,000 feet long, is to discharge 24,000 gallons of water per hour; what must be the head?

SOLUTION.— $\dfrac{24,000}{60 \times 60} = 6\frac{2}{3}$ gallons per second.   Using formula **185**

$$v_m = \frac{24.51\,Q}{d^2} = \frac{24.51 \times 6\frac{2}{3}}{16} = 10.2 \text{ feet per second.}$$

In Table 45, $f = .0205$ for $v_m = 8$, and .0193 for $v_m = 12$.

$.0205 - .0193 = .0012 =$ difference for a difference in the mean velocities of 4 ft. per sec. $.0012 \div 4 = .0003 =$ difference for a difference of 1 ft. per sec. $10.2 - 8 = 2.2$. $.0003 \times 2.2 = .00066$. $.0205 - .00066 = .01984$.   Then, substituting in formula **183**,

$$h = \frac{f\,l\,v_m^2}{5.36\,d} + .0233\,v_m^2 = \frac{.01984 \times 2,000 \times 10.2^2}{5.36 \times 4} + .0233 \times 10.2^2 = 195 \text{ ft}$$

Ans.

BENDS AND ELBOWS.

2211.   In laying pipes, all bends and elbows should be avoided as much as possible.   When they are absolutely necessary, they should be as large as the circumstances will

permit, so as to change the direction gradually.   Sudden
changes in direction destroy the velocity very rapidly, and,
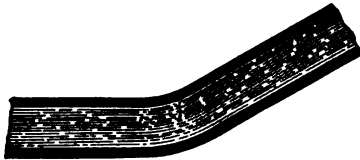
FIG. 750.
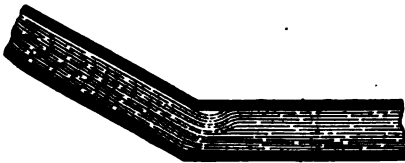
FIG. 751.
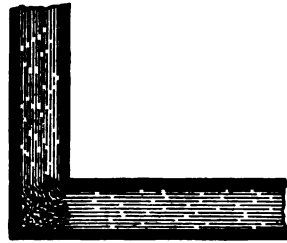
FIG. 752.

consequently, reduce the discharge.   A reduction or in-
crease in the size of the pipe, owing to the screwing on of
branch pipes smaller or larger than the main pipe, also
reduces the velocity.

When bends are necessary,
it is better to round them, as
shown in Fig. 750, than to have
a sharp bend, as shown in Fig.
751.   A bend at right angles,
as shown in Fig. 752, is very
destructive to the velocity.   A
rounded elbow, as shown in
Fig. 753, should be used, in
which the radius should be made as large as possible.
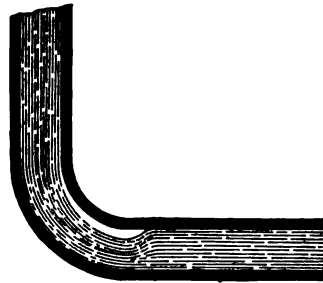
FIG. 753.

PUMPS.

DESCRIPTION OF TYPES.

2212. The Suction-Pump.—A section of an ordi-
nary suction-pump is shown in Fig. 754.   Suppose the piston
to be at the bottom of the cylinder, and to be just on the
point of moving upwards in the direction of the arrow.   As

the piston rises, it leaves a vacuum behind it; the atmospheric pressure upon the surface of the water in the well causes it to rise in the pipe *P*, for the same reason that the mercury rises in the barometer-tube. The water rushes up the pipe, and lifts the valve *V*, filling the empty space in the cylinder *B*, displaced by the piston. When the piston has reached the end of its stroke, the water entirely fills the space between the bottom of the piston and the bottom of the cylinder, and also the pipe *P*. The instant that the piston begins its down stroke,
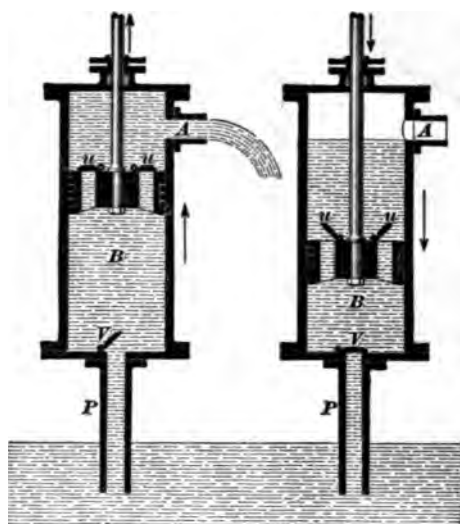
FIG. 754.

the water in the chamber *B* tends to fall back into the well, and its weight forces the valve *V* to its seat, thus preventing any downward flow of the water. The piston now tends to compress the water in the chamber *B*; but this is prevented by the opening of the valves *u, u* in the piston. When the piston has reached the end of its downward stroke, the weight of the water above closes the valves *u, u*. All the water resting on the top of the piston is then lifted with the piston on its upward stroke, and discharged through the spout *A*, the valve *V* again opening, and the water filling the space below the piston as before.

It is evident that the distance between the valve *V* and the surface of the water in the well must not exceed 34 feet, the highest column of water which the pressure of the atmosphere will sustain, since, otherwise, the water in the pipe would not reach to the height of the valve *V*. In practice, this distance should not exceed 28 feet, or, to obtain the best

ffects, not more than 22 feet.   This is due to the fact that
here is a little air left between the bottom of the piston
nd the bottom of the cylinder; a little air leaks through
he valves, which are not perfectly air-tight, and a pressure
; needed to raise the valve against its weight, which, of
ourse, acts downwards.   There are many varieties of the
iction-pump, differing principally in the valves and piston;
ut the principle is the same in all.

2213.   The Lifting-Pump.—A section of a lifting-
ump is shown in Fig. 755.   These pumps are used when
ater is to be raised to greater heights
ian can be done with the ordinary
iction-pump.   As will be perceived,
; is essentially the same as the pump
reviously described, except that the
out is fitted with a cock, and has a
ipe attached to it, leading to the
oint of discharge.   If it is desired to
scharge the water at the spout, the
ck may be opened; otherwise, the
ck is closed, and the water is lifted
· the piston up through the pipe P'
the point of discharge, the valve c
eventing it from falling back into
e pump, and the valve V preventing
e water in the pump from falling
ck into the well.   It is not necessary
it there should be a second pipe P',
shown in the figure, for the pipe P
y be continued straight upwards.
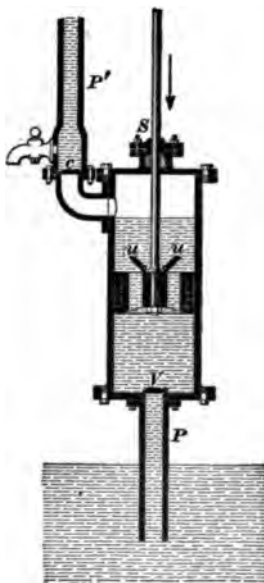
FIG. 755.

[n all pumps, the pipe that conducts the water or other
uid to the pump-cylinder is called the **suction-pipe ;** the
e that conducts the water away from the pump-cylinder
:alled the **delivery** or **discharge pipe.**   In Figs. 755
1 756, P is the suction and P' the delivery pipe.   The
:tion-pipe is sometimes called the **inlet-pipe.**

2214. Force-Pumps.—The force-pump differs from the lifting-pump in several important particulars, but chiefly in the fact that the piston is solid; that is, it has no valves. A section of a suction and force pump is shown in Fig. 756. The water is drawn up the suction-pipe *P*, as before, when the piston rises; but, when the piston reverses, the pressure on the water caused by the descent of the piston opens the valve *V''*, and *forces* the water up the delivery-pipe *P'*. When the piston again begins its upward movement, the valve *V''* is closed by the pressure of the water above it, and the valve *V* is opened by the pressure of the atmosphere on the water below it, as in the previous cases. For an arrangement of this kind, it is not necessary to have a stuffing-box. The water may be forced to almost any desired height. The force-pump differs again from the lifting-pump in respect to its piston-rod, which should not be longer than is absolutely necessary in order to prevent it from *buckling*, while in the lifting-pump the length of the piston-rod is a matter of indifference.
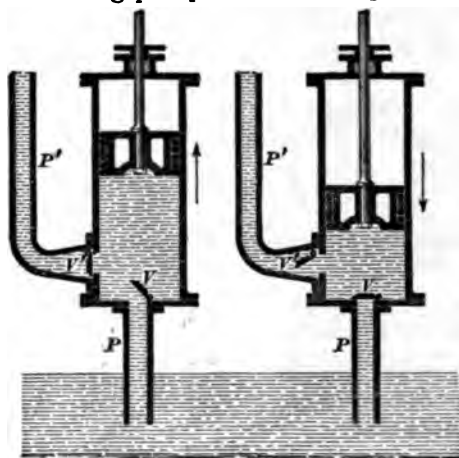
FIG. 756.

2215. Double-Acting Pumps.—In the pumps previously described, the discharge was intermittent; that is the pump could only discharge when the piston was moving in one direction. In some cases, it is necessary that there should be a continuous discharge; in all cases, it takes more power to run the pump with an intermittent discharge, as a little consideration will show. If the height to which the water is to be raised is considerable, its weight will be very

great, and the entire mass must be put in motion during one stroke of the piston.

In order to obtain the advantage of a more continuous discharge, double-acting pumps are used. Fig. 757 shows a part sectional view of such a pump. Two pistons a and b are used, which are operated by one handle c, in the manner
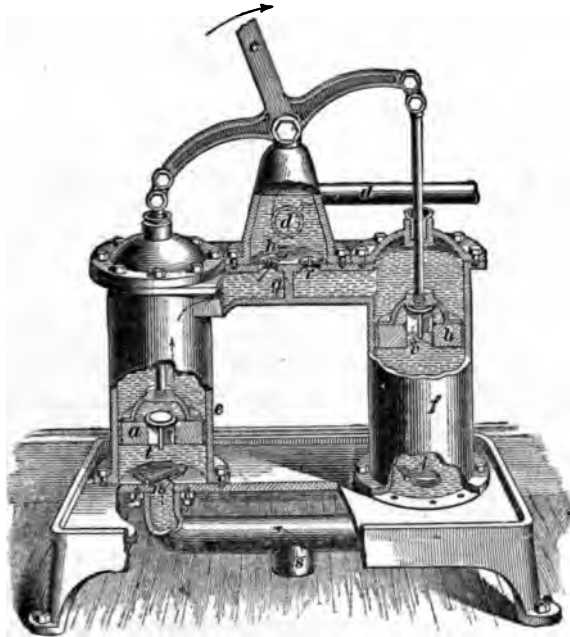
FIG. 757.

shown. The pump has one suction-pipe s and one discharge-pipe d. The cylinders e and f are separated by a diaphragm g, so that they can not communicate with each other above the pistons. In the figure, the handle c is moving to the right, the piston a upwards, and the piston b downwards. As the piston a moves upwards, it lifts the water above it, and causes it to flow through the delivery-valve h into the discharge-pipe d. This upward movement of the piston creates a partial vacuum below it in the cylinder e, and causes the water to rush up the suction-pipe s into the

24—20

cylinder, as shown by the arrows. In the cylinder f, the down-ward movement of the piston b raises the piston-valve v, and the weight of the water on the suction-valve i keeps it closed. When the handle c has completed its movement to the right, and begins its return, all the valves on the right-hand side open except v, and those on the left-hand side close except t; water is then discharged into the delivery-pipe by the cylinder f, and only at the instant of reversal is the flow into the delivery-pipe d stopped.

2216. Air-Chambers.—In order to obtain a con-tinuous flow of water in the delivery-pipe, with as nearly a uniform velocity as possible, an **air-chamber** is usually placed on the delivery-pipe of force-pumps as near to the pump-cylinder as the construction of the machine will allow. The air-chambers are usually pear-shaped, with the small end connected to the pipe. They are filled with air which the water compresses during the discharge. During the suction, the air thus compressed expands, and acts as an accelerating force upon the moving column of water, a force which diminishes with the expansion of the air, and helps to keep the velocity of the moving column more nearly uniform. An air-chamber is sometimes placed upon the suction-pipe. These air-chambers not only tend to pro-mote a uniform discharge, but also to equalize the stresses upon the pump, and prevent shocks due to the incompressi-bility of water. They serve the same purpose in pumps that a fly-wheel does to the steam-engine. Unless the pump moves very slowly, it is absolutely necessary to have an air-chamber on the delivery-pipe.

2217. Steam-Pumps.—Steam-pumps are force-pumps operated by steam acting upon the piston of a steam-engine directly connected to the pump, and, in many cases, cast with the pump. A section of a double-acting steam-pump showing the steam and water cylinders, with other details, is illustrated in Fig. 758. Here G is the steam-piston, and R the piston-rod, which is secured at its other end to the pump-plunger P. F is a partition cast with the

cylinder, which prevents the water in the left-hand half from communicating with that in the right-hand half of the cylinder.   Suppose the piston to be moving in the direction indicated by the arrow.   The volume of the left-hand half of the pump-cylinder will be increased by an amount equal to the area of the circumference of the plunger, multiplied by the length of the stroke, and the volume of the right-hand half of the cylinder will be diminished by a like amount.   In consequence of this, a volume of water in the
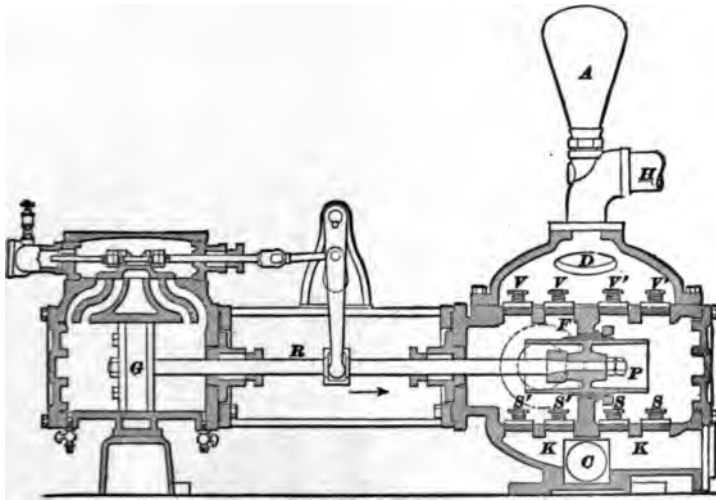
FIG. 758.

right-hand half of the cylinder equal to the volume displaced by the plunger in its forward movement will be forced through the valves V', V' into the air-chamber A, through the orifice D, and then discharged through the delivery-pipe H.   By reason of the partial vacuum in the left-hand half of the pump-cylinder, owing to this movement of the plunger, the water will be drawn from the reservoir through the suction-pipe C into the chamber $K K$, lifting the valves S', S', and filling the space displaced by the plunger.   During the return stroke, the water will be drawn through the valves S, S into the right-hand half of the pump-cylinder, and discharged through the valves V, V

in the left-hand half.   Each one of the four suction and four discharge valves is kept to its seat when not working by light springs, as shown.

There are many varieties and makes of steam-pumps, the majority of which are double-acting.   In many cases, two steam-pumps are placed side by side, having a common delivery-pipe.   This arrangement is called a **duplex pump.**   It is usual to so set the steam-pistons of duplex pumps, that when one is completing the stroke, the other is in the middle of its stroke.   A double-acting duplex pump made to run in this manner, and having an air-chamber of sufficient size, will deliver water with nearly a uniform velocity.

In mine-pumps for forcing water to great heights, the plungers are made solid, and in most cases extend through the pump-cylinder.   In many steam-pumps, pistons are used instead of plungers; but, when very heavy duty is required, plungers are to be preferred.

2218.   Centrifugal Pumps.—Next to the direct-acting steam-pump, the centrifugal pump is the most valuable instrument for raising water to great heights that has yet been described.   As the name denotes, the effects produced by centrifugal force are made use of.   Fig. 759 represents one with half of the casing removed. The hub S is hollow, and is connected directly to the suction-pipe.   The curved arms a, called **vanes,** or **wings,** are revolved with a high velocity in the direction indicated by the arrow, and the air enclosed between them is driven out through the discharge-
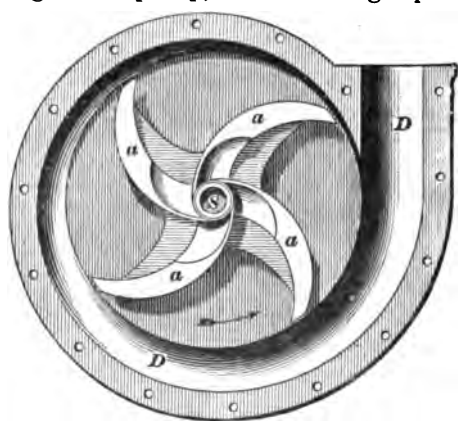
FIG. 759.

ssage and delivery-pipe $D D$. This creates a partial
cuum in the casing and suction-pipe, and causes the water
flow in through S. This water is also made to revolve
th the vanes, and, of course, with the same velocity. The
itrifugal force of the revolving water causes it to fly out-
rds towards the end of the vanes, and becomes greater
: farther away it gets from the center. This causes it to
ve the vanes, and, finally, to leave the pump by means of
: discharge-passage and delivery-pipe $D D$. The height
which the water can be forced depends upon the velocity
the revolving vanes. In the construction of the centrif-
al pump, particular care is exercised in giving the correct
m to the vanes; the efficiency of the machine depends
atly upon this point being attended to. What is re-
red is to raise the water; and the energy used to drive
 pump should be devoted as far as possible to this one
pose. The water, when it is raised, should be delivered
h as little velocity as possible; for any velocity which the
er then possesses has been produced at the expense of
energy used to drive the pump. The form of the vanes
uch that the velocity with which the water leaves the
ip is reduced to an amount just sufficient for its delivery
he proper height.

he number of vanes depends upon the size and capacity
he pump. It will be noticed that in the pump shown in
figure, the vanes have sharp edges near the hub. The
:ct of this is to provide for a free ingress of the water,
 also to cut any foreign substance that may enter the
ip, and prevent it from working properly.

entrifugal pumps are sometimes used to raise water
 feet or more, but they work much more economically
 heights of 25 to 40 feet. Almost any liquid can be
ed with these pumps; but, when used for pumping chem-
s, the casing and vanes are made of materials that the
nicals will not affect.

iud, gravel, etc., can also be raised when mixed with
er, the wear due to these impurities being very slight.

POWER REQUIRED FOR PUMPS.

2219. Rule.—In all pumps, whether lifting, force, steam, single or double-acting, or centrifugal, *the number of foot-pounds needed to work the pump is equal to the weight of the water in pounds, multiplied by the vertical distance in feet between the level of the water in the well, or source, and the point of discharge, plus the work necessary to overcome the friction and other resistances.*

2220. Rule.—*The work done in one stroke of a pump is equal to the weight of a volume of water equal to the volume displaced by the piston during the stroke, multiplied by the total vertical distance in feet through which the water is to be raised, plus the work necessary to overcome the resistances.*

2221. A little consideration will make this evident. Suppose that the height of the suction is 25 feet; that the vertical distance between the suction-valve and the point of discharge is 100 feet; that the stroke of the piston is 15 inches, and its diameter is 10 inches. Let the diameters of the suction-pipe and delivery-pipe be 4 inches each. The volume displaced by the pump piston or plunger in one stroke equals

$$\frac{10^2 \times .7854 \times 15}{1,728} = .68177 \text{ cubic foot.}$$

The weight of an equal volume of water $= .68177 \times 62.5 = 42.61063$ pounds. Now, in order to discharge this water, *all* of the water in the suction and delivery pipes had to be moved through a certain distance in feet equal to .68177, divided by the area of the pipes in square feet.

4 inches $= \frac{1}{3}$ of a foot. $(\frac{1}{3})^2 \times .7854 = \dfrac{.7854}{9} = .0872\frac{2}{3}$ sq. ft. $.68177 \div .0872\frac{2}{3} = 7.8125$ feet.

The weight of the water in the delivery-pipe is $(\frac{1}{3})^2 \times .7854 \times 100 \times 62.5 = 545.42$ pounds.

The weight of the water in the suction-pipe is $(\frac{1}{3})^2 \times .7854 \times 25 \times 62.5 = 136.35$ pounds.

$545.42 + 136.35 = 681.77$ pounds $=$ the total weight of water moved in one stroke. The distance that it is moved

in one stroke is 7.8125 feet. Hence, the number of foot-pounds necessary for one stroke is $681.77 \times 7.8125 = 5,326.33$ foot-pounds. Had this result been obtained by the rule given in Art. **2220,** the process would have been as follows: The weight of the water displaced by the piston in one stroke was found to be 42.61063 pounds. $42.61 \times 125 = 5,326.33$ pounds, which is exactly the same as the result obtained by the previous method, and is a great deal shorter.

EXAMPLE.—What must be the necessary horsepower of a double-acting steam-pump, if the vertical distance between the point of discharge and the point of suction is 96 feet? The diameter of the pump-cylinder is 8 inches; the stroke is 10 inches, and the number of strokes per minute is 120. Add ⅓ for friction, etc.

SOLUTION.—Since the pump is double-acting, it raises a quantity of water equal to the volume displaced by the plunger at every stroke. The weight of the volume of water displaced in one stroke $= (\frac{8}{12})^2 \times$ $.7854 \times \frac{10}{12} \times 62.5 = 18.18$ lb., nearly.

$18.18 \times 96 \times 120 = 209,433.6$ foot-pounds per minute.

Since ⅓ is to be added for friction, etc., the actual number of foot-pounds per minute is $209,433.6 + \dfrac{209,433.6}{3} = 279,244.8$.

One horsepower $= 33,000$ foot-pounds per minute; hence, $\dfrac{279,244.8}{33,000} =$ 8.462 H. P., nearly. Ans.

DUTY OF A PUMP.

2222. *The* **duty** *of any pump or pumping-engine is the number of pounds of water raised one foot high for each 100 pounds of coal burned in the boiler.*

2223. The duty is calculated by multiplying the number of pounds of water discharged in one hour by 100, and by the total height in feet that the water is lifted, and dividing the product by the number of pounds of coal burned during the hour. Since the discharge is usually given in gallons, the following formula may be used, in which

$G =$ number of gallons discharged per hour;

$h =$ total vertical distance in feet between the level of the water in sump, or other source of supply, and the point of discharge;

$W =$ weight in pounds of coal burned per hour;
$D =$ duty in foot-pounds.

$$D = \frac{835.5\, G\, h}{W}. \qquad \textbf{(187.)}$$

EXAMPLE.—What is the duty of a pump whose discharge is 88,152 gallons per hour, the height of lift being 630 feet, and the number of pounds of coal burned per hour being 580?

SOLUTION.—Applying formula **187**,

$$D = \frac{835.5\, G\, h}{W} = \frac{835.5 \times 88,152 \times 630}{580} = 80,000,000 \text{ foot-pounds, nearly.}$$
<div align="right">Ans.</div>

2224. If every unit of heat developed by the combustion of 100 pounds of coal of average composition could be converted into the work of raising water without any loss due to friction, leakage, etc., the duty developed would be between 1,100,000,000 and 1,200,000,000 foot-pounds. Knowing this to be a fact, the duty of a pump is a measure of its actual efficiency. The highest duty test on record, in which the engines used were of the triple-expansion type, and every precaution taken to insure the best coal being used, gave as the result a duty of about 143,000,000 foot-pounds, about ⅛ of the full value of the coal. This is considered a remarkable result.

MINE-PUMPING MACHINERY.

2225. Cornish Pumping-Engines.—Until within comparatively recent times, the so-called **Cornish pumping-engines** have been the only ones used for removing the water from the mines. This engine was invented by Watt for use in the mines of Cornwall, and was the first really effective steam-engine made. An illustration of a Cornish pumping-engine is shown in Fig. 760. The cylinder A is single-acting; that is, the steam acts only on one side of the piston. Its diameter is, in this case, 70 inches, and its stroke is 10 feet. B is the piston-rod; it is connected to the walking-beam C by a link R. In Cornish pumping-engines, the steam is admitted, through the valve in I, to the top of the piston, and forces it down towards the

bottom of the cylinder.   The weight of the pump-rods and other moving parts in the shaft, called the **pit-work**, is
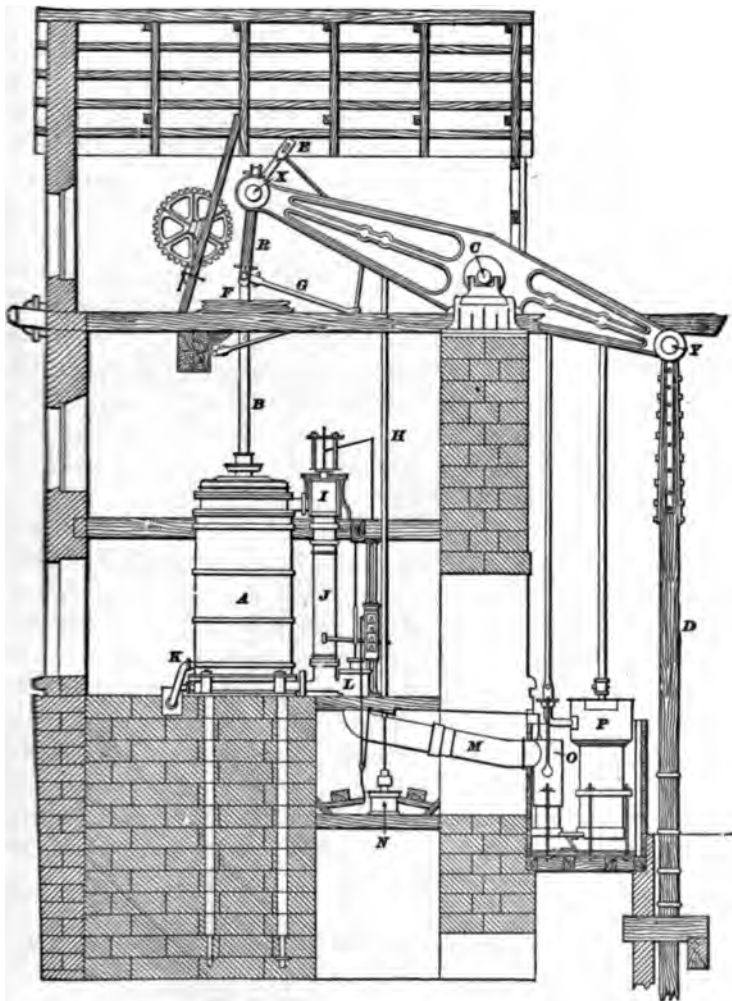
FIG. 760

sufficient to raise the piston to the top of the cylinder when the steam on the upper side of the piston is put in communication with the lower side.   The cylinder A is steam-jacketed; that is, the cylinder-walls are hollow, and filled

with steam in a manner similar to the water-jacket of a
air-compressor, the steam entering through the pipe *K*.

The action of the pump is as follows: Steam is admitte
to the upper side of the piston through a valve in *I*, whic
is operated by means of a tappet-rod *H*. The steam is
high pressure, and forces the piston-rod downwards, and :
the same time raises the pit-work. This gathers momentu
while coming upwards, and the steam is cut off, expandii
during the rest of the stroke. Just before the end of t
stroke, what is termed an equilibrium-valve, also located
the casing at *I*, opens, and allows the steam in the upper e
of the cylinder to communicate with that in the lower en
The two pressures being thus balanced, the heavy pit-wc
causes the right end of the walking-beam *C* to descer
raising the piston to the top of the cylinder again. T
exhaust-valve is located at *L*. When this is raised, the
haust steam flows through the pipe *M* into the condenser
P is a small pump used in operating the condenser. *E* is
catch, intended to act in case the valves should fail to wol
The piston-rod passes between two blocks, of which *F* is ol
the other being opposite. If the left end of the walkir
beam should descend too far, a cross-piece on the catch-r
E is caught by the blocks *F*, and prevents any further dow
ward movement of the piston. It will be noticed that t
left end of the beam, which turns on the journal *C*, is long
than the right end. This is made thus in order to have t
stroke of the pumps less than the stroke of the steam-cyli
der, which, in many cases, is longer than it is desired
have the pump-stroke. If *s* represents the length of tl
piston-stroke, the pump-stroke will be $s \times \dfrac{C\,Y}{X\,C}$, when $X\,C$ ar
C Y represent the lengths of the two arms of the beam :
shown in the cut.

2226. Cornish Bull Engine.—In Fig. 761 are shov
two views of a Cornish Bull engine and pump. This sty
of pumping-engine is made by many firms, and differs bi
very little in regard to details. Here the walking-beam
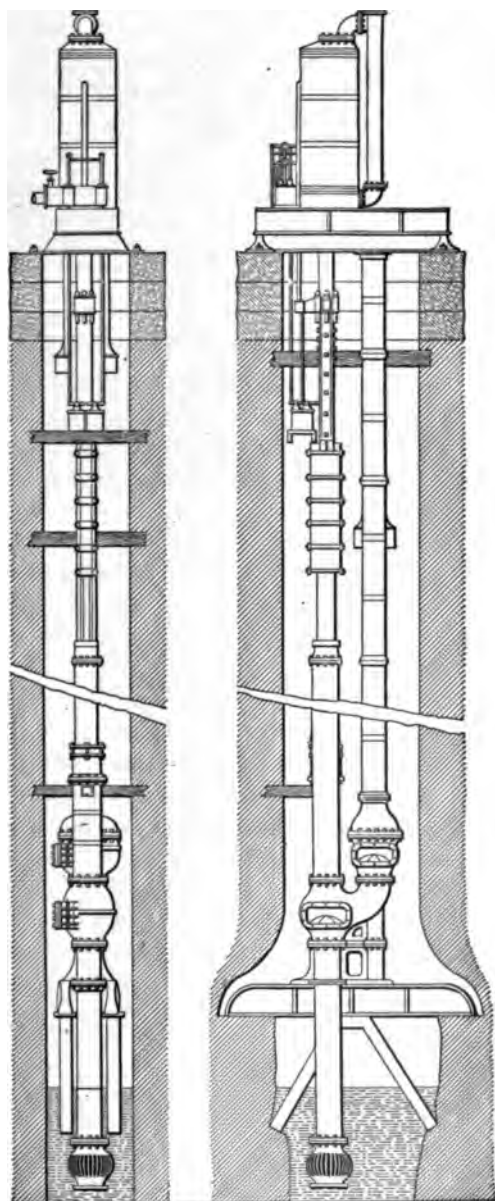
FIG. 761.

dispensed with, and the cylinder is placed directly over the shaft, the pit-work being attached to the piston itself. In this case also the cylinder is single-acting, the steam being admitted below the piston instead of above it, as in the engine described in Fig. 760. The condenser is usually omitted in this class of pumps, the steam exhausting directly into the atmosphere. The pump shown is for a single lift; for depths greater than about 350 feet, the lift is generally in two or more operations, which will be described later. In case the weight of the pit-work should be greater than necessary to force the water up the required height, the extra weight is counterbalanced in a manner to be described later.

The Bull pumping-engine possesses several advantages over the Cornish pump. The heavy walking-beam and its connections are dispensed with; this lessens the first cost; the friction is greatly reduced; the advantage of having a direct-acting engine is also obtained. The principal disadvantage is that, the pump being directly over the shaft, it takes up a great deal more room, where space is necessary, than the Cornish pump.

Cornish and Bull pumps both use steam expansively. They do not have fly-wheels to absorb the energy of the early part of the stroke and give it out again at the end, but utilize the heavy pit-work to accomplish the same purpose. The number of expansions ranges from four to ten; that is, the steam is cut off from $\frac{1}{4}$ to $\frac{1}{10}$ stroke. When using more than 6 expansions ($\frac{1}{6}$ cut-off), the strain produced on the machinery becomes very heavy, and the resulting wear and tear of the machinery more than makes up for the increased economy in the use of steam. Many engineers claim that four expansions are the most economical.

2227. In the following table, the sizes of a number of Bull engines and pumps in use in the Wyoming coal-fields are given. The number of gallons discharged, given in this table, is the actual quantity delivered.

ENGINES.

neter of nder.	Length of Stroke.	No. of Strokes per Minute.	Diameter of Steam-Pipe.	Diameter of Exhaust-Pipe.
in.	9 ft.	15	6 in.	8 in.
in.	10 ft.	12	8 in.	10 in.
in.	10 ft.	10	9 in.	12 in.
in.	10 ft.	8	10 in.	14 in.

PUMPS.

ter er.	Stroke of Plunger.	Diameter of Suction-Pipe.	Diameter of Discharge-Pipe.	Gallons per Minute.	Height of Lift.
l.	9 ft.	20 in.	20 in.	2,203	84 ft.
l.	10 ft.	21 in.	21 in.	1,958	300 ft.
l.	10 ft.	23 in.	23 in.	1,975	{ 600 ft., 2 lifts.
l.	10 ft.	25 in.	25 in.	1,880	{ 833 ft., 2 lifts.

28. In nearly every case where surface pumps are
ɔ raise water from great depths, plunger (force) pumps
nployed, the weight of the pit-work, which includes
:ight of the plungers, being sufficient to overcome the
: of the water and the friction attending its motion.

29. In Fig. 762 is shown a section of a lifting-pump
e in mines. The pump consists of a series of pipes
:ted together, of which the lower end only is shown in
t. That part of the pipe included between the letters
l *B* forms the pump-cylinder, in which the piston *P*
    The part above the highest point of the piston
is the delivery-pipe, and the part below the lowest

point of the piston travel is the s⌐
pipe. When speaking of these pa
applied to mine-pumps, the delivei
is usually termed the **working b**
and the suction-pipe the **wind-bore**

In all cases of mine-pumps, the low
of the wind-bore is pear-shaped, an
forated with many small holes, to kee
matter in the water from enterir
pump and destroying the valves. Iı
cases, the pear-shaped end is covere⌐
gauze for the same purpose. *C* is ,
covering in the pipe, which may
moved to allow the suction-valve
repaired. *D* is another plate cove
similar opening, to allow the pisto
piston-valves to be repaired. The ⌐
rod, or, rather, the piston-stem, is
of wrought iron, inserted with woo
connected to the piston in the n
shown by the cut. The only limit
height to which a pump of this kiı
raise water is the strength of the ⌐
rod. The pipe continues straigł
wards in the direction of the arrow.

The piston-rod is raised by means
engine situated at the surface—ei
Cornish engine, Bull engine, or an
nary horizontal steam-engine—opera
walking-beam in a manner hereafter
described.

2230. The lifting-pump shown i
755, where a separate pipe is used t
vey the water from the pump-cylin
the point of discharge, requires the
rod to be round, and to pass thro
stuffing-box, as shown, in order to p⌐

FIG. 762.

(a)

(c)

(b)
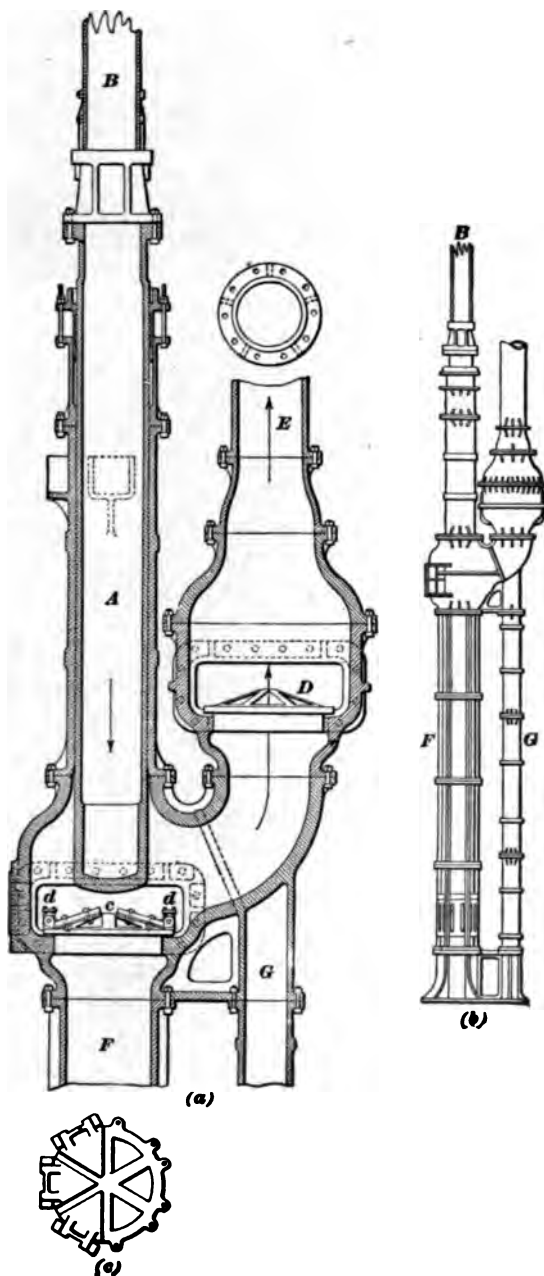
FIG. 762.

leakage. The pump shown in Fig. 762 may have a rod of
any shape, and no stuffing-box is required.

2231. In Fig. 763 is shown a **plunger-pump** (force
pump) used in connection with the Bull engine shown i·
Fig. 761; (*b*) is an elevation of the pump, and (*a*) is a sec·
tion drawn to a larger scale, showing the pump-cylinder and
valves. As will be seen, the plunger A is hollow, the weight
of the heavy rod B and connections being sufficient to rai s
the water to the required height. The method of attaching
the rod to the plunger is fully shown by the cut. Suppose
the plunger to be on the down (indoor) stroke; the valve c
is, of course, closed; the water filling the pump-cylinder *is*
forced through the valve D, which it opens, and up the de-
livery-pipe E. When the plunger reaches the end of *its*
stroke and begins its return, the weight of the water forces
the valve D to its seat, retaining the water above it in the
discharge-pipe E. As the plunger moves upwards, it leaves
a partial vacuum behind it, causing the water to rush up the
suction-pipe F, lift the valve c, and fill the pump-cylinder.
The plunger makes another downward stroke, and the above
process is repeated. G is a standard attached to the deliv-
ery-pipe, the lower end resting on a foundation. This is
necessary, since the great weight of the water in the dis-
charge-pipe and the weight of the pipe itself would break it
off at the bend, unless supported in some such manner;
otherwise the thickness of the metal around the bend would
necessarily be enormous.

A top view of the valves is shown at (*c*). They con-
sist of six triangular valves arranged in a circle, with their
apexes pointing towards the center. These six valves turn
upwards on hinges, and are prevented from going too
far by the projection *d*; see (*a*). Three of the valves
have been removed so as to show the amount of valve-
opening which they give. When the valves are open, they
form an angle of about 45° with their position when closed.

**2232. Advantages and Disadvantages of Lift-
Pumps.**—It is easier to specify the objections to lift-pumps

than to state their advantages over the plunger-pumps. The pump-rod, being necessarily inside of the delivery-pipe, reduces the effective area of pipe, and increases the friction of the water to some extent, owing to the added surface rubbed against. The rods are concealed, and can not be inspected without removing the entire rod. Not only do the bolts and rods sometimes break, thus rendering their recovery difficult, but the bolts will wear against the stocks, causing loss of power by friction and destroying the pipes.

Lift-pumps are not so liable to sudden injurious strains as the plunger-pumps. They are better adapted for sinking purposes than the plunger-pumps, the impurities in the water being less harmful than in the case of plunger-pumps.

The plunger type of pumps is superior to the lift-pump in nearly every respect for very high lifts, with the accompanying heavy pressure, or when dirty water is being raised. When pumping against a heavy pressure, it is impossible to keep the piston of lift-pumps tight, and prevent the water from leaking. The piston and cylinder of the lift-pump must in every case be a perfect fit, and be truly cylindrical. With plunger-pumps, on the contrary, the rod passes through a stuffing-box, and the plunger may or may not fit the cylinder. When pumping dirty water, the grit comes in contact with the surface that the piston of a lift-pump is constantly traveling over, and destroys both the cylinder and piston very rapidly; whereas, the plunger has to be kept tight at only one permanent place, and the dirt cannot very well get at the surface of the packing on which the plunger or plunger-rod rubs. Every part of a plunger-pump can be readily examined and repaired without taking down the whole apparatus.

PUMP DETAILS.

VALVES.

2233. A section of a pump clack-valve is shown in Fig. 764. *A* and *B* are the clacks, lined with leather on the bottom, to make a tighter fit on the seat, and thus do away with the necessity of grinding the valve when fitting. *C* is

a small casting for the clacks to strike against, so as to prevent them from opening too far, and E is the pin or axle on which they turn. D is a cylindrical casing having a flange around the outside near the middle.   The upper part of this casing forms the valve-seat.   These valves are of the
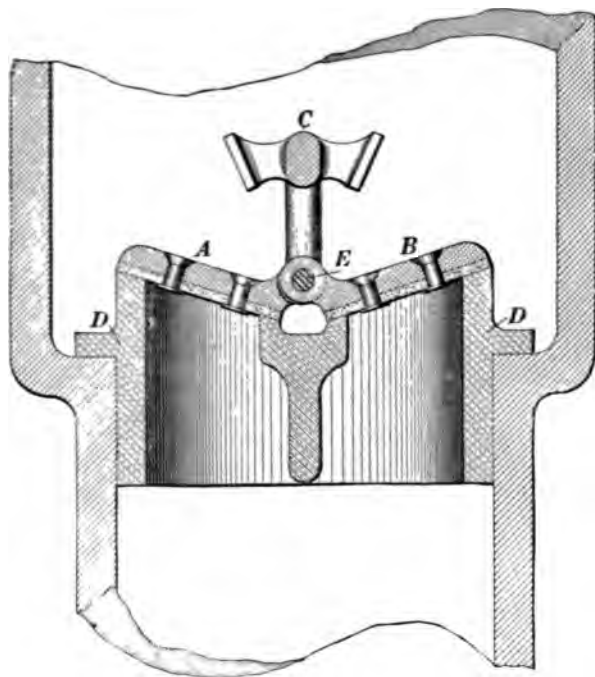
FIG. 764.

type known as the **butterfly valve,** and are used more than any other form for pumps at collieries, principally because of their cheapness, simplicity of construction, and the fact that they can be set up, repaired, or replaced by any one who is handy with tools.

2234.   Fig. 765 shows a section of what is termed a **single-beat valve.** A is the valve; B is a stem which forms a part of the back of the valve and acts as a guide in the bearing D. C, C, C, C are rubber rings, kept in position between the flange of the fixed bearing D and the

back of the valve *A* by means of the stem, and separated by the washers *E, E, E.* When the water raises the valve, the rings are compressed, and the shock which would be produced by the valve striking the flange is done away with, and with it the liability of breaking the valve. The rings likewise assist in closing the valve. This is called a single-beat valve,
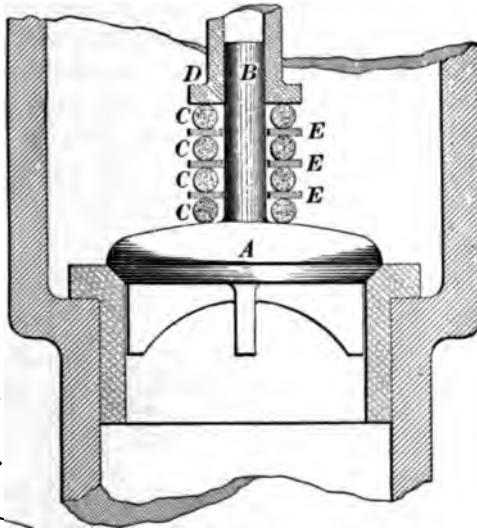
FIG. 765.

because there is but one opening. These valves are used for lifts up to 500 feet.
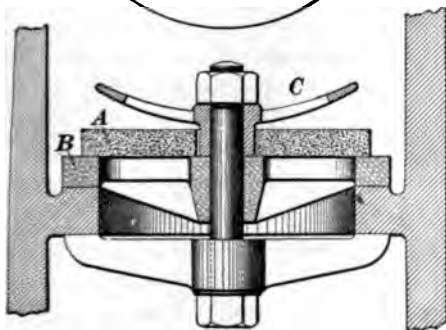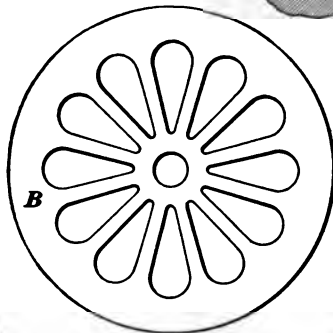
2235. For lifts up to 300 feet, India-rubber disk valves give good satisfaction. Fig. 766 shows one of ordinary construction. An India-rubber disk *A* is fixed over the center of the grid *B*. When the water rushes through the holes, the rubber disk is lifted at its ends until it strikes the curved piece *C*, which

FIG. 766.

is placed there to keep the valve from opening too far. The grid, or seat, *B* is filled with holes, through which the water passes. This valve works well, but is not satisfactory on account of the necessity of constantly renewing it. The valve can not turn, and, therefore, rises and falls back into the same position every time. In consequence of this, the heavy water causes those portions of the rubber disks which cover the holes in the seat to be
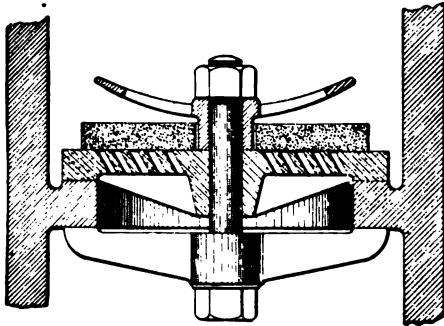
FIG. 767.

pressed inwards, and the valves wear out very fast. To obviate t h i s difficulty, the holes in the grid are slanted, as shown in Fig. 767, and a small brass collar is fixed to the center of the disk. The water, rushing in at an angle, rotates the valve slightly each time, thus presenting a new surface to the holes in the seat, and reducing the wear to a minimum. In a later construction of this valve, the grid passages are vertical, as in Fig. 766, but the disk itself has inclined teeth cut in the circumference. This answers the same purpose as the method shown in Fig. 767, and is superior to it, since the direction of the water is not changed by inclined openings.

2236. A section of a **Cornish double-beat** valve is shown in Fig. 768. This valve is deservedly a favorite, and is used when high pressures are required. Besides being used for a water-valve, it may be used for steam or air. These valves have been applied to pumps working against a head of 700 feet with entirely satisfactory results. They are called double-beat valves because they have two seats and two openings for discharge. *A* is the casing which slides on the vertical stem *B*; when down, it rests on the valve-seats at *C* and *D*. When the pressure below becomes

ter than that above, it raises the casing, and the water
lischarged through the circular openings at C and D.
: rib around the outside of the casing is for the purpose
strengthening it.   The valve-seats are conical.   The
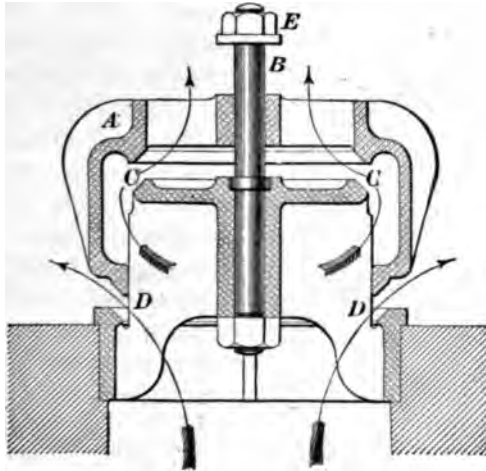
FIG. 768.

re shows that one opening discharges the water outside
:he valve, and the other through the inside.   In some
s the valve-seats are flat instead of conical, and have a
> of rubber extending around the entire seat.

PIPES.

237. The pipes used to convey the water from the
: to the surface are generally made in sections of about
'eet in length, with flanges on each end to bolt them
ther.   The usual practice is to make them of cast iron;
ι the water is not injurious, however, they are some-
s made of wrought iron, on account of the great reduction
eight; wrought iron being so much stronger than cast
the thickness of the pipe may be a great deal less.

number of different forms of joints are used, one of
est being shown in Fig. 769.   The projection B (some-
s called a *spigot*) is made just strong enough to pre-
its being broken when connecting up the pipes; its

purpose is to keep the pipes in line. *C* is a ring of lead, or wrought iron, from $\frac{1}{8}$ to $\frac{1}{4}$ inch thick; its purpose is to make a tight joint, and it is wrapped with flannel, or common woolen cloth, soaked in tar.   The lead ring is preferred on account of the wrought-iron one having a decided tendency to corrode. When wrought-iron r i n g s a r e used, the pipes must be faced, b u t   t h i s is not necessary w h e n   lead is employed, owing to its softness.   In all first-class work the pipe-flanges are faced.   The bolts are distributed around the flange, as shown in the plan.
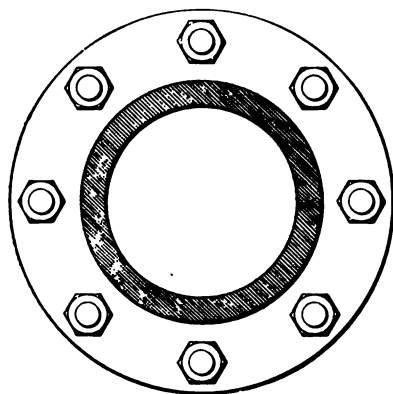
2238. A s o m e w h a t similar joint is shown in
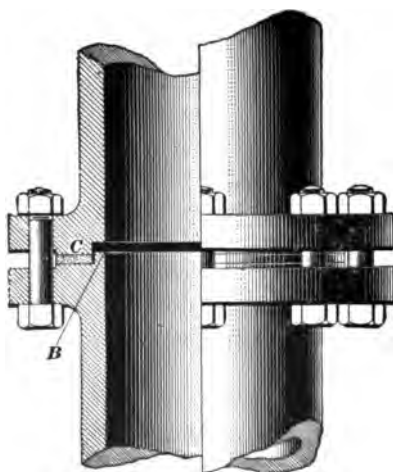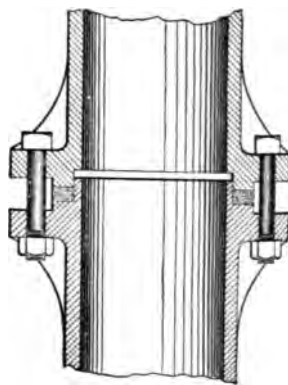
FIG. 769.   FIG. 770.

Fig. 770.   Here the flange is strengthened by heavy ribs, to prevent its being broken when tightening the nuts.

2239. A very good joint, which will withstand pressures up to 1,000 pounds per square inch, is shown in

Fig. 771. It consists of two ordinary flanges, faced straight across, having a triangular groove cut in each. A gutta-percha cord is made into a ring by cutting its ends beveling, and making the cord of just such a length as will equal the circumference of the **V** groove. The cord is then laid in the groove with the two ends matching, and when the nuts are screwed down, the cord spreads, filling the entire groove as shown. The greater the pressure in the pipe, the greater the stress on the flat surface of the triangular ring; consequently, the greater the compression of the gutta-percha, the better the joint.
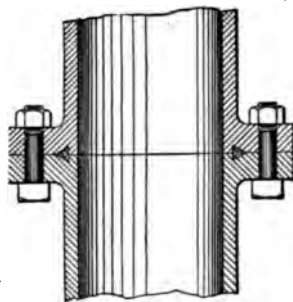
FIG. 771.

ROD-JOINTS.

2240. A method of connecting the different sections of the pump-rod (also called *spear-rod*) is shown in Fig. 772. Here the two ends of the rod are squared, and placed butt to butt at *A*. Four wrought-iron plates are then bolted on, the bolts passing through the rod. Some colliery engineers object to four plates, claiming that the extra bolts reduce the strength of the rod too much; hence they use two plates. Four plates stiffen the rod a great deal more than two, and, since the rods are usually made a great deal larger than absolute strength requires, it is better to have four plates than two. These plates, called **strapping-plates,** are usually made thick in the middle, where the joint comes, and then tapering in both directions to half that thickness at the ends. Many engineers make them of equal thickness the whole length, in order to save the extra cost of forging.

FIG. 772.

FIG. 773.   FIG. 774.   FIG. 775.

2241. Fig. 773 shows two views of a joint in which cast-iron strapping-plates are used. In this case, the bolts do not pass through the rod, but to one side. Two plates are used, which have rectangular-shaped lugs *A*, on the bottom, between which the wooden part of the rod fits exactly. The two parts of the rod are held together by the friction between the rods and the plates. This is an excellent joint, and is to be commended, when properly made.

2242. In Fig. 774 is shown another excellent joint, which must be made with great care. The two ends of the pieces forming the joint are cut like *A* and *C*. They are placed together, and the two strapping-plates put on similarly to the method shown in Fig. 772. Then a square pin *B* is driven in, and in the best construction two other plates are bolted to the other two sides. This is a costly joint, on account of the great care necessary to make the two pieces fit. It is a good plan, however, to use it when practicable. All pieces forming the rod should be of equal length, in order that one piece may be readily taken out and replaced by a duplicate, when repairs are necessary. A fair length to allow would be thirty feet. They should be longer, rather than shorter than this, in order to reduce the number of joints, but it would be well not to exceed forty-five feet. In all cases, the joints should be so made that there will be no "lost motion"; that is, that there shall not be any space between the two ends joined together, so that they will be liable to wear and have a little end-play when the direction of the stroke is reversed. All such lost motion shortens the stroke of the engine, and lessens the discharge, besides increasing the wear and tear.

2243. When several plungers are operated by one rod, as is usually the case with deep shafts, they are connected to the rod by means of an **off-set**, see Fig. 775. The cut shows three views of the most common method adopted for uniting the plunger-pole and the main pump-rod. *D* is the plunger-pole; *E* the main pump-rod; *F* the off-set; *B*, *B* are staples made of round iron bars, on which a screw-thread

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is cut at each end; *A*, *A* are the glands or cross-bars, shown in the plan, against which the nuts are tightened.   .

CATCHES.

2244.   In order to prevent the pump-rods from falling down the shaft, in case of the valve-gear refusing to work, and thus allowing the piston to blow through the cylinder, a catch is located at the top of the shaft.   Fig. 776 shows
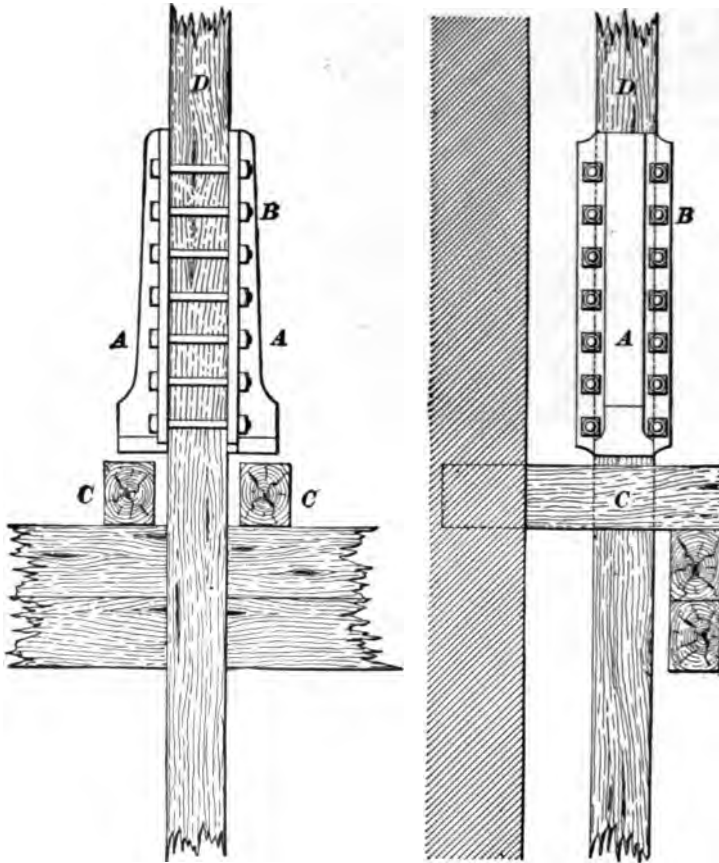
FIG. 776.

the arrangement of one of these catches.   It consists of two cast-iron pieces *A*, *A*, attached to the pump-rod *D* by

means of bolts. These bolts do not pass through the rod, but are situated on the sides, very close to it, so as not to weaken the rod. *C, C* are two banging-pieces, one end of which is carried to the sides of the shaft, the other end being secured to two beams which are stretched across the shaft, and between which the rod moves. The catch and banging-pieces are so located that the piston can not touch the cylinder-head in any case.

Another method is to secure two wooden pieces on each side of the rod by means of bands having screw-threads cut on the ends, in much the same manner that the off-set was arranged in Fig. 775. When the shaft is very deep, and there are several plungers operated by one rod, it is usual to have a catch near each plunger.

GUIDING THE RODS.

2245. The usual manner of guiding the rods is illustrated in Fig. 777. Two beams *A, A* are placed across the
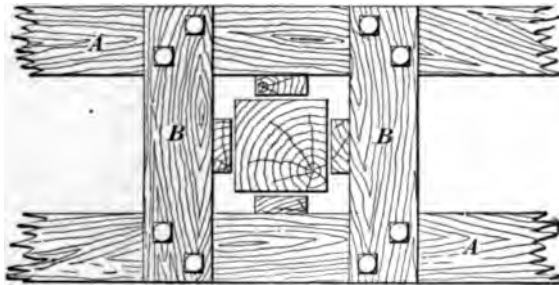
FIG. 777.

shaft, and two crosspieces *B, B* are attached to them. On each of these four pieces is a rubbing-block, the rod sliding loosely between them.

GENERAL ARRANGEMENT.

2246. The entire arrangement of shaft, pump-rods, engines, counterweights, etc., is shown in Fig. 778. Instead of a Cornish or Bull engine, as has been heretofore described, an ordinary horizontal engine is used.

32 ft.10 in.

1640 ft.

295.2 ft.

328 ft.

328 ft.

328 ft.

360.8 ft.
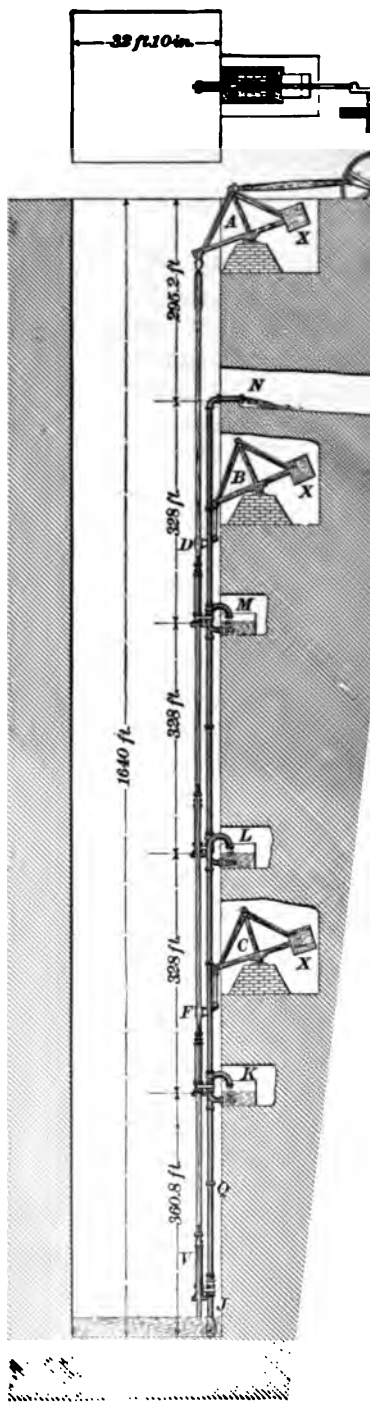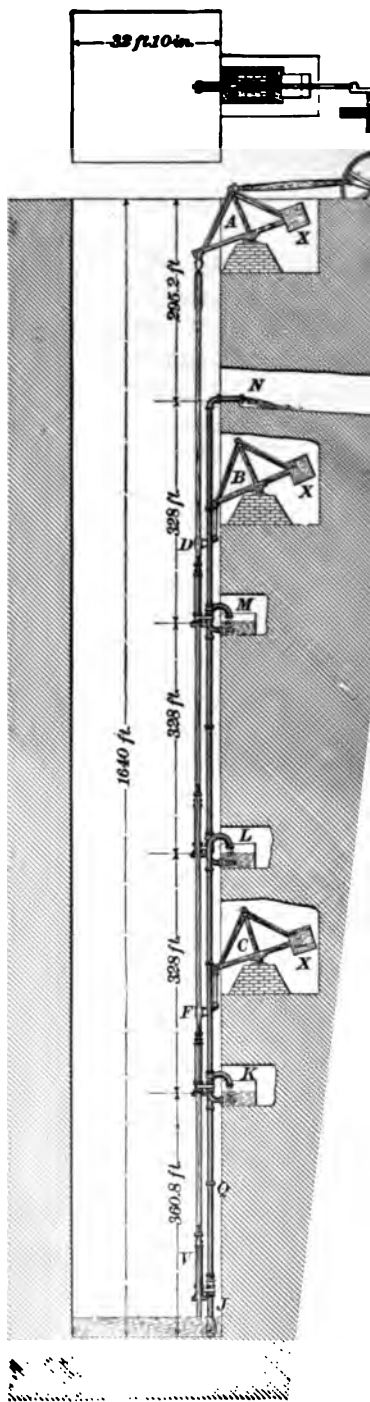
The type of engine employed in this case is a **Corliss-geared engine.** Fig. 779 shows in greater detail an engine of different make, but the same in principle as the one shown in Fig. 778, and which is used for the same purpose. In an arrangement of this kind, it is necessary that the pump-rod should move very slowly while the steam economy is increased by higher speeds. Hence, in Fig. 779, the crank-shaft has keyed to it a stepped pinion, which meshes into a very large stepped gear-wheel. A second crank is keyed to the shaft of the large gear-wheel, and to it is attached one end of a long wooden connecting-rod T, whose other end is attached to a bell-crank in a manner shown more clearly in Fig. 778. The second crank thus communicates its motion to the bell-crank A, Fig. 778, which in turn operates the pump-rod in the shaft.
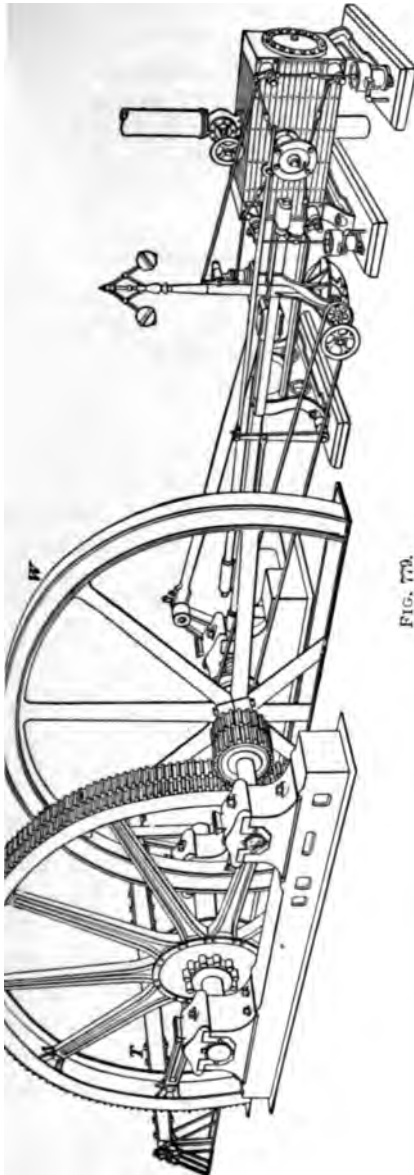
FIG. 779.

On account of the great length of the rod (over 1,600 feet), its weight added to the weight of the plungers is considerably more than the weight of the water-column; hence, to save the extra power which would be required to be used in raising this extra weight, it is counterbalanced.   A counterbalance-weight X is placed on one end of the bell-crank A; two other bell-cranks, B and C, are located down the shaft, one end carrying the counterweight X and the other end being connected to the pump-rod by means of a link and the cast-iron off-sets D and F.

The shaft itself is square, one side measuring 32 ft. 10 in. The water is raised by four lifts, the first, to K, being 360.8 ft., and the other three 328 ft. each.   In this particular instance, the water is discharged into a tunnel N, about 300 feet below the surface.   The pump-rod goes straight down the shaft, and the discharge-pipes are placed alternately on each side of it. J is a suction-pipe; I is a bracket, one end of which is attached to the pump-rod and the other end to the pump-plunger V.   On the down stroke, the water is forced out of the pump-cylinders and up the pipes Q, R, S, and U, discharging at K, L, M, and N.   The discharge-pipe is 8 inches in diameter.

This pumping arrangement possesses several advantages over the Cornish or Bull pumping-engines.   The fly-wheel permits a more even distribution of the power.   The length of the stroke is always the same, and there is no danger of damage caused by the piston being blown through the cylinder-head, should the valve-gear refuse to work.

The crank, instead of being made as shown, may be made in the form of a circular disk, and have several pins on it at varying distances from the center, so that the stroke of the engine may be lengthened or shortened, if desired.   This, however, is not recommended.   The engine can also run at a great deal higher speed, while the pump-rods move no faster than the Cornish type.   The gears increase the engine friction to some extent, but the loss from this source is probably no more than in the case of the walking-beam engine.

BALANCING THE PUMP-RODS.

2247. It has been stated that the water is forced upwards by the weight of the descending pit-work. The weight of the pit-work must then, of course, be greater than the weight of the ascending column of water, and the velocity of the descending pit-work will depend directly on the difference between its weight and the weight of the water-column. There is, however, a practical limit which the speed of the pit-work may not exceed; viz., about 200 feet per minute, or less. One reason for this limit is the liability of the piston to pound the cylinder-head, if moving too fast; and another is that the velocity of the water in the pipe should be not more than 200 to 250 ft. per minute. If, then, the difference between the weight of the pit-work and of the water-column be too large for the required velocity, a balance-bob must be resorted to, as shown in Fig. 778. The pump-rod, in descending, has not only to raise the water-column, but also to lift the weight at the end of the bell-crank. The speed of the descending pit-work can thus be exactly regulated by the weight of the balance.

2248. As an example, suppose the weight of the pit-work is 20 tons, the weight of the water-column raised is 16 tons, and the frictional resistances, say, 3 tons. First find the velocity of the pit-work on the down stroke, and see if the pump-rod needs to be balanced. The total force is 20 tons, and the total resistance is $16 + 3 = 19$ tons, leaving a net force of 1 ton to accelerate the moving mass. The weight to be accelerated is $20 + 16 = 36$ tons; the friction, of course, not requiring acceleration. The formula which expresses the relation between the force, acceleration, and weight, is

$$f = \frac{g\,F}{W}, \qquad (\mathbf{188.})$$

where F is the force; W, the weight; f, the acceleration in feet per second, and g, the acceleration due to gravity, which is usually taken as 32.16 ft. per sec. in a second.

Substituting the values of F, W, and g, in formula **188**,

$$f = \frac{32.16 \times 1}{36} = .89\tfrac{1}{3} \text{ ft. per sec. in a second.}$$

That is, *the velocity increases regularly at the rate of .89¼ ft. per sec.*

2249. Assuming the stroke of the engine to be 10 ft., the time of one stroke, when the piston has the acceleration given above, is found by the following formula:

$$t = \sqrt{\frac{2\,s}{f}}, \qquad \textbf{(189.)}$$

in which t is the time in seconds, and s is the space passed over in feet.   Substituting, $t = \sqrt{\dfrac{2 \times 10}{.89\frac{1}{4}}} = 4.732$ seconds.

Ten feet in 4.732 seconds is at the rate of $\dfrac{10}{4.732} \times 60 = 127$ ft. per min., nearly, which is well within the required velocity, and, therefore, no balance is needed.   In fact, the pump-rod might advantageously be weighted a little, and its speed thereby increased.

Suppose, however, that instead of 20 tons the pit-work had weighed 24 tons, the other conditions remaining the same. The net force free to produce acceleration would then be $24 - 19 = 5$ tons, and the weight to be accelerated would be $24 + 16 = 40$ tons.   Using formula **188,**

$$f = \frac{g\,F}{H} = \frac{32.16 \times 5}{40} = 4.02 \text{ ft. per sec.}$$

Substituting this value of f in formula **189,**

$$t = \sqrt{\frac{2\,s}{f}} = \sqrt{\frac{2 \times 10}{4.02}} = 2.23 \text{ sec., nearly.}$$

Since the stroke of 10 feet is made in 2.23 seconds, the average velocity per minute is $\dfrac{10}{2.23} \times 60 = 269$ ft. per minute. As this speed is rather too high, it should be reduced by means of a counterweight.   Suppose that a counterweight of two tons be tried.   The weight of the pit-work is, as before, 24 tons; the weight which it puts in motion is 16 tons (weight of water) plus 2 (counterweight) = 18 tons. $24 - 18 = 6$ tons = force available to move the water, coun-

terweight, and to overcome the frictional resistances. Since the total weights have been increased from $16 + 20 = 36$ tons, originally, to $24 + 18 = 42$ tons, the frictional resistances have also been increased. Assuming them to be $3\frac{1}{2}$ tons now, the effective force left to produce acceleration is $6 - 3.5 = 2.5$ tons $= F$. Substituting in formula **188,**

$$f = \frac{32.16 \times 2.5}{42} = 1.9143 \text{ ft. per sec.}$$

Substituting this value of f in formula **189,**

$$t = \sqrt{\frac{2 \times 10}{1.9143}} = 3.2323 \text{ sec.}$$

Consequently, the speed in feet per minute is

$$\frac{10}{3.2323} \times 60 = 186 \text{ ft. per min., nearly.}$$

This is a fair speed, but should a higher rate be required, all that will be necessary will be to reduce the counterweight.

UNDERGROUND MINE-PUMPS.

2250. **Underground direct-acting mine-pumps** may be **simple** or **compound,** and either may be of the **single** or **duplex** type. They may be run by steam or compressed air, and the simple pumps (single or duplex) may also be run by water or electricity. There are many different makes of these pumps, which, like the steam-engine, differ more or less in their details, the principle governing each type being the same in all. In the following pages a description of one pump of each type will be given.

2251. In mine-pumps, plungers are almost invariably used instead of pistons. A **simple Worthington single pump** was shown in Fig. 758. By **single pump** is meant a pump which has but one pump-cylinder in contradistinction to the **duplex,** which has two, and the **triplex,** which has three, pump-cylinders. The words **simple** and **compound** refer to the steam-cylinder. Hence, a pump may be a **simple single** or a **simple duplex,** a **compound single** or a **compound duplex,** etc.

24—22

2252. The pump shown in Fig. 758 is, as mentioned before, a **simple single steam-pump.** When the water pumped is gritty and brings extraordinary wear upon the plunger and bushing that it slides in, which is attached to the partition F, a stuffing-box is placed at F.

The manner of attaching the stuffing-box is shown in Fig. 780. In this case the valve arrangement (not shown) is altered somewhat. A pump whose plunger is packed in this
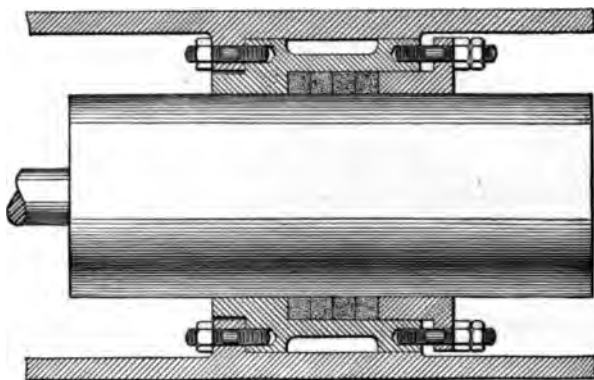
FIG. 780.

manner is called an **inside-packed pump.** Consequently, the pump shown in Fig. 758 would be called a **simple inside-packed single pump.** The great disadvantage in the use of this pump is that the water must be drained from the cylinder and the cylinder-head removed in order to repair the packing. In order to remove this difficulty, the so-called **outside-packed plunger-pumps** are used.

Fig. 781 shows a **simple outside-packed single mine-pump.** Two plungers F and F' are connected by the yokes H and rods I, on each side, so that both plungers move at the same time. The plunger F is attached directly to the piston-rod P. Suppose the steam-piston in D to be moving in the direction indicated by the arrow; the plunger F is then forcing water into the chamber G, and up the delivery-pipe J. The discharge-valve in the chamber G' is closed, and the plunger F', being forced outwards by the yokes H

ind rods *I*, leaves a vacuum behind it, which is filled by
water from the suction-pipe.   On the return stroke, the
above operations are reversed, *F'* doing the forcing and *F*
the suction.   It should be understood that the cylinders in
which *F* and *F'* work are separated by a partition at *N*, in
somewhat the same manner as the two halves of the cylinder
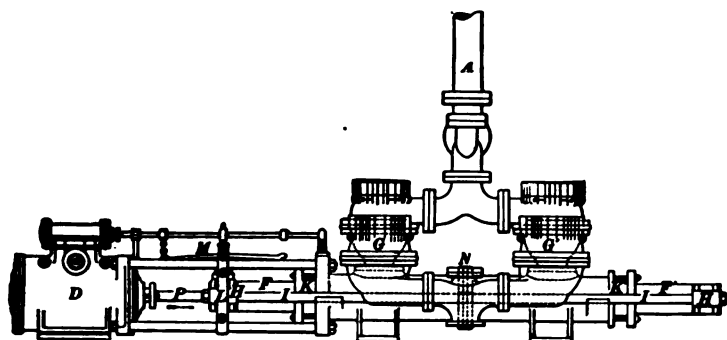
FIG. 781.

in Fig. 758.   A detailed description of this arrangement
applied to a pump of different manufacture will be described
further on.

Two stuffing-boxes, *K* and *K'*, are used to pack the
plungers.   As will be seen, they are outside of the cylinders,
and the bushing can be easily removed, the packing repaired,
and the bushing replaced without disturbing the cylinder-
head itself in the slightest.   The steam-valve is operated in
this case by means of the lever *M*, carried to and fro by the
upright-piece *L*, which is attached to the piston-rod *P*.

For most purposes, the outside-packed mine-pump is
superior to the inside-packed type, but takes up more room.

2253.   Fig. 782 shows a **Worthington simple out-
side-packed duplex mine-pump.**   The plunger *A* has
nearly completed its stroke in the direction indicated by
the arrow, while the plunger *B* has completed the same por-
tion of its stroke in the opposite direction.   The steam-valve
in the chamber *D* is operated by means of the crank *F*, act-
ing through the long bearing *G*, and actuating the crank *H*,
attached to the valve-stem by a link, as shown.   The valve

in the chamber E is actuated in a similar manner by means of the crank I.   These cranks are so set that the plungers A and B are always moving in opposite directions.   This arrangement, in combination with the air-chamber L, produces a nearly uniform discharge.   Both pump-cylinders dis-
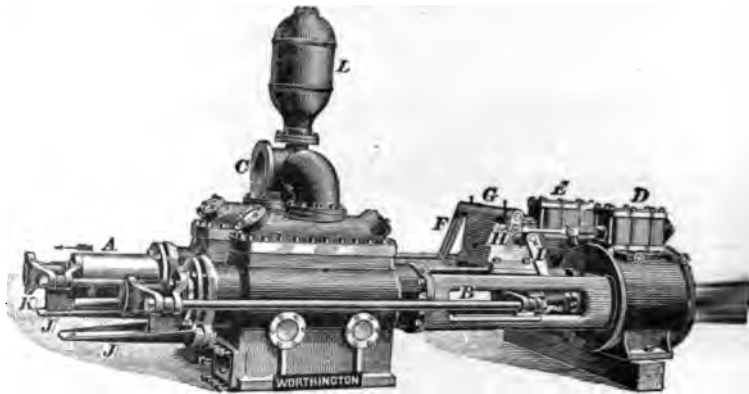
FIG. 782.

charge into the same delivery-pipe C. K is a *shoe*, one being attached to both ends of both plungers, and moves on the slide J at the left, and on the cross-head slide at the right. They prevent the ends of the plungers from falling downwards out of line when near the end of the stroke.   The shoes are adjustable for wear.   The ends of the plungers are connected by means of yokes and rods in a manner similar to the pump last described.   As will be seen, it is outside-packed.

The chief disadvantage of this valve-driving arrangement is that one pump can not be disconnected from the other when the work would require only one pump to be in use. In order that one pump may run, the other must also be in motion.

2254.   In Fig. 783 is shown a perspective view of a **Jeanesville compound outside-packed duplex mine-pump.**   This is a very powerful pump, the one illustrated having the following dimensions:

Diameter of high-pressure cylinder......25 inches.
Diameter of low-pressure cylinder ......42 inches.
Diameter of pump-plungers ....14 inches.
Stroke of plungers and pistons..........48 inches.

Its rated capacity is 2,000 gallons of water per minute
;ainst a head of 425 feet.   In compound pumps, the steam
carried full stroke in both cylinders, the expansion being
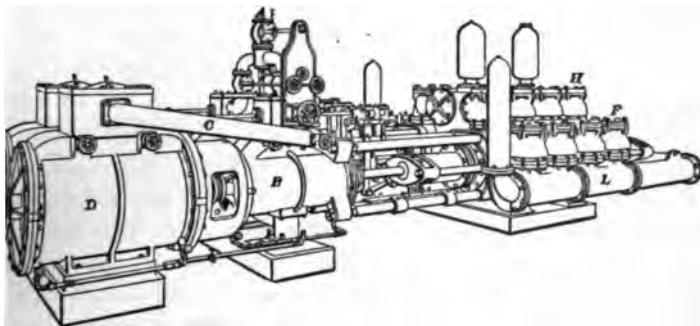
FIG. 783.

tained through the difference of cylinder volumes.
nce the stroke is the same in both cylinders in the pump
own in the figure, the volumes will be to each other as the
uares of the diameters, or

lume of low-pressure cylinder : volume of high-pressure
cylinder :: 42² : 25².

$$42^2 = 1,764 ; \quad 25^2 = 625; \quad \frac{1,764}{625} = 2.82, \text{ nearly.}$$

Hence, the volume of the low-pressure cylinder is, in this
stance, 2.82 times that of the high-pressure cylinder, and
e steam expands 2.82 times.

As this is a duplex pump, there are two high-pressure and
o low-pressure cylinders. A is the throttle-valve; B the
gh-pressure cylinder of one pump; C the steam-pipe
lich connects one high with one low pressure cylinder,
d D the low-pressure cylinder.   The other pump is exactly
nilar.   The throttle-valve A admits steam to both high-
essure cylinders.

2255. In order to show more clearly the working
the various parts of this pump, a sectional view is shown
Fig. 784. Here, A is the main throttle-valve; B is
auxiliary throttle-valve leading from the main valve to c
of the high-pressure cylinders, a similar one being plac
on the other side of the main valve connecting with t
other high-pressure cylinder. The object of these auxilia
valves is to allow more or less steam to be admitted to c
side in case it should appear to be doing less or more wc
than the other, the pump working better when both si
are doing the same amount of work. C is the piston-rod,
which are connected both pistons and the plunger E. 1
figure shows that the plungers are hollow. The form of t
shoe K, which supports the ends of the plungers, can
clearly seen. The shape of the slide J on the back ei
upon which the shoe moves, is indicated by dotted lin
G is the partition or diaphragm which separates the t
plunger-cylinders. All so-called **double-plunger** pun
require this diaphragm, so that when the plungers i
moving towards it, the water can find no way of esca
except through the delivery-valves. L is the sucti
pipe.

2256. Referring now to both Figs. 783 and 784, it v
be noticed that there are eight valves for both sets of pl
gers, four suction-valves F, and four discharge-valves
making 16 valves in all. The usual practice is to hav
large number of small valves, from 50 to 100, for a pu
of this size, instead of a small number of large valv
The mine water in the anthracite coal-fields attacks i
only iron, but brass and phosphor-bronze as well, mak
the life of a valve-seating a limited one at best. This f
makes it imperative that pump-valves for mines be simp
strong, easy of examination, and quick of replacement.
is certainly easier to take care of 8 or 16 large valves th
50 to 100 small ones. Then, too, the parts of a large va
are heavier, and are less liable to be twisted or brok
owing to rough handling.

2257. A section through two of the discharge-valves is shown in Fig. 784. In order to more clearly show the working of these valves, an enlarged view is given in Fig.
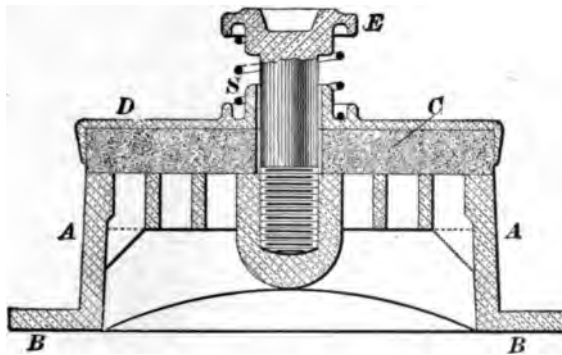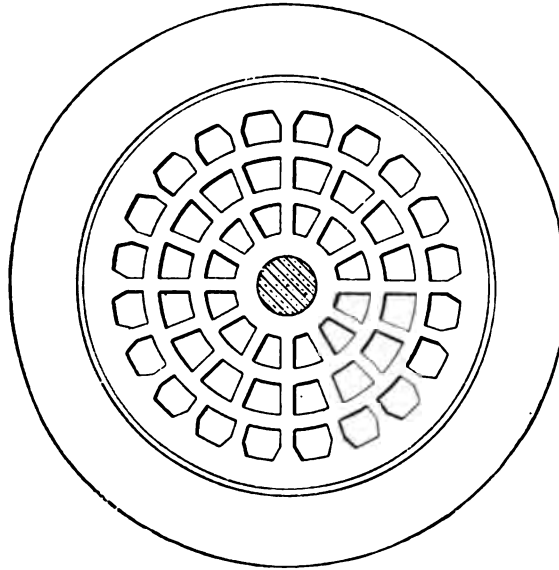
FIG. 785.

785. The valve-seat A is held in place by means of the flange B (see also Fig. 784). As shown in the top view, the valve-seat is perforated by a large number of small holes; this is necessary, since the valve C is made of rubber. The

cap D is made of gun-metal; the spring S and bolt E of phosphor-bronze.

Another pump of this description and of the same make is in actual use near Wilkes-Barre, Pa., working against a head of 1,060 feet.   It is a **22′ and 36′ × 9′ × 36′ compound duplex;** that is, the high-pressure cylinder, low-pressure cylinder, and plungers are respectively 22′, 36′, and 9′ in diameter, and the stroke is 36′.

2258.   Fig. 786 shows a **Knowles compound condensing outside-packed duplex mine-pump.**   The arrangement of the high and low pressure cylinders and of the plunger-cylinders is similar in all respects to the pump described in Fig. 783, and, hence, will not be repeated here.
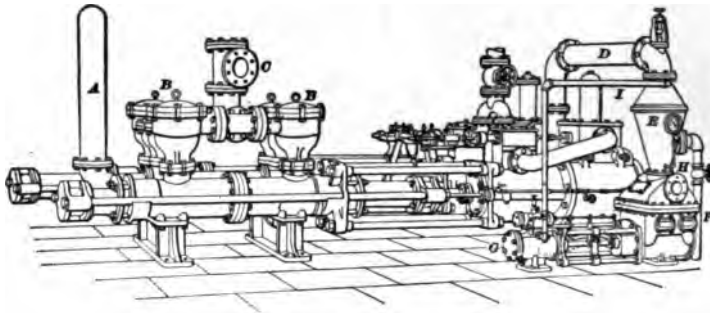
FIG. 786.

There are four discharge-valves—one for each plunger-cylinder.   The suction-valves are not seen in this view. A is one end of the suction-pipe, the other end being in the sump from which the water is taken.   This pipe lies between the two sets of water-cylinders, and has communication with all of the four plunger-cylinders.   The discharge-valves B are what the makers term " **pot-valves** "; they are claimed to be of great durability, and work on independent gun-metal composition seats.   Both valves and seats can be easily taken out and examined. C is the flange to which the discharge-pipe is bolted.   After the steam has been used in the low-pressure cylinder, it is discharged into a condenser E through the pipe D.   After being condensed, the water

is discharged through a pipe bolted to the flange H. F is
the pipe which leads the cold water to the condensing-chamber E. G is the air-pump for removing the condensed
water and discharging it either into the boiler, if placed
near the pump, or into the sump. I is a pipe through which
the exhaust steam from the air-pump passes to the condenser.

These pumps are intended for very heavy duty, the one
shown in the cut being designed for a discharge of 1,000 gallons per minute under a head of 800 feet. They will pump
water vertically 1,000 feet or over on single lifts.

SINKING-PUMPS.

2259. When putting down a new shaft or deepening
an old one, the so-called **sinking-pump** is used to drain
the water from the shaft-bottom, so that the work may proceed. These pumps must necessarily be portable, and are
suspended by a chain attached to eye-bolts in the pump.
They are also provided with wrought-iron clamps, by means
of which they may be attached to the timbers in the shaft
when it is desired to fix them in position temporarily.
Hence, as the shaft gets deeper, the chain may be lengthened out, an extra joint placed on the upper end of the
delivery-pipe, and it is again ready for business. The sinking-pump is subjected to the hardest usage of any other
mine machine. The water pumped is invariably gritty and
often acid. The water trickling down on the pump from
above carries mud along with it, and so completely covers
the pump that it is hardly distinguishable at times from the
soil itself. Notwithstanding all this, a sinking-pump must
work night and day, often up to the limit of its capacity, and
its failure, even for a day, at a critical period, may flood a
shaft, which would require a week or more to recover.

2260. In Fig. 787 is illustrated two views of a **Cameron sinking-pump.** This pump meets all of the conditions required of a sinking-pump, and is a favorite with
mine operators. There is no outside valve mechanism
whatever, and nothing short of actual breakage of the pump

itself, or of the steam, suction, or delivery pipe, can pre-
vent the pump from working.   The manner of suspending
it from a chain is shown in the cut, also the method of at-
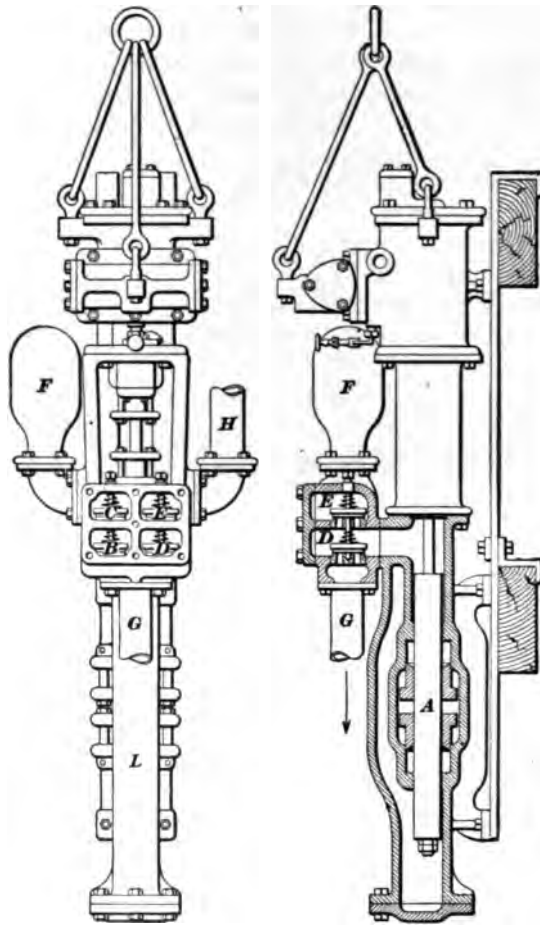
FIG. 787.

taching it to the shaft-timbers.   In order to more clearly
show the working of the valves and plunger, a partial section
of the pump is given in the figure.   The pump has one
plunger, and is double-acting.   Instead of employing a

phragm to separate the two plunger-cylinders, as in the
np illustrated in Fig. 784, two stuffing-boxes are placed
he center to accomplish the same purpose.   This device
requently used in ordinary horizontal mine-pumps, and
:n so used the pumps are called **center-packed,** to dis-
ļuish them from the **inside-packed** and **outside-**
:ked pumps.   The center-packed pump is considered
erior to the inside-packed pump for mining purposes, but
to be so convenient as the outside-packed pump.   The
ter-packed sinking-pump, however, is considered superior
the other two.   The action of this pump is as follows:
⸮ is the suction-pipe and H the delivery-pipe.   Suppose
: plunger to be moving in the direction indicated by
: arrow.   The water is forced out of the chamber L,
ich communicates with the delivery-pipe H by means
the valve C, and lifts C, thus flowing into H.   As
: plunger moves down it leaves a vacuum behind it; the
ter in the shaft rushes up the suction-pipe, raises the
ve D, and fills the upper part of the plunger-cylinder.
ien the stroke is reversed, the valves C and D close, and
: valves E and B open, the water being forced up the
e H, through the valve E, and the chamber L is filled
ough the opening of the valve B. F is, of course, the
·chamber.   The section shown by the view on the right
aken in a rather peculiar manner, the greater part being
en through the center line of the engine, so as to show
plunger, stuffing-boxes, etc., and the part showing the
ves being taken on the center line of the valves E and D
the view on the left.
t is quite customary to use a sinking-pump to raise the
er from the sump to the first station, since the sinking-
ıp may be raised or lowered according to the depth of
water in the sump.   A single steam-pipe down the
ft supplies both the sinking-pump and the main pump.
en used for this purpose, the sinking-pump exhausts into
sump.
ın account of its portability, the sinking-pump ıs espe-
ly adapted to the recovery of flooded mines..

LOCATION OF PUMPS.

2261. Fig. 788 shows a partial section of a mine. It is intended to illustrate the different positions of the pump, pipes, sump, etc. *A* is the sump, or a reservoir filled by means of a sinking-pump raising the water to *A* from a sump below, and *B* is a compound condensing duplex pump. *C* is the condenser and air-pump. *D* is the steam-pipe which is carried down the shaft to the pump. *D'* is a smaller steam-pipe, leading from the main pipe *D* to the air-pump which it drives. *E* is the pipe which connects the condenser
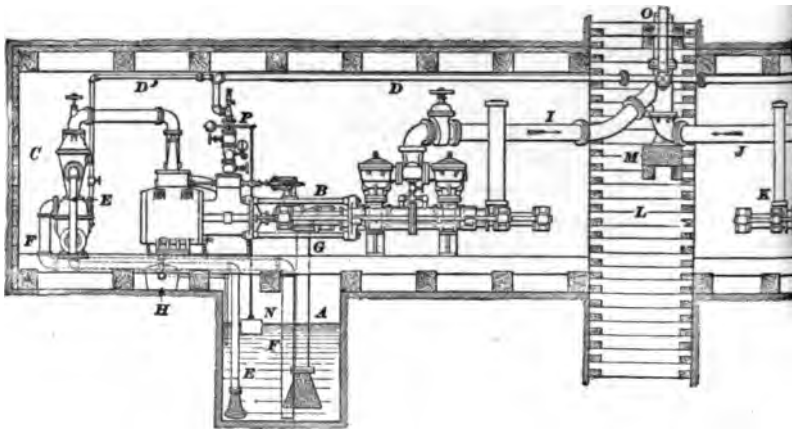
FIG. 788.

with the sump; it supplies the cold water needed for condensing the steam. This water is conveyed to the condenser on the same principle that the water flows through the suction-pipe of a pump. The steam is condensed, and leaves a partial vacuum in the condensing-chamber; the atmospheric pressure forces the water in the sump up the pipe *E*. After the steam delivered by one stroke of the pump has been condensed, the condensed steam, together with the water used to condense it, is pumped back into the sump through the pipe *F*. It is evident that the colder the injection water, the better will the condenser perform its duty. Consequently, in order to enable the pipe *E* to obtain as cold water as possible, the pipe *F* discharges very

se to the pump suction-pipe *G*.   *H* is a steam-trap for
: purpose of removing entrained water and water of con-
ısation from the steam before entering the pump.   *I* is·
 delivery-pipe which connects directly with the column·
e *O* (main delivery-pipe) in the shaft *L*.   *N* is a float
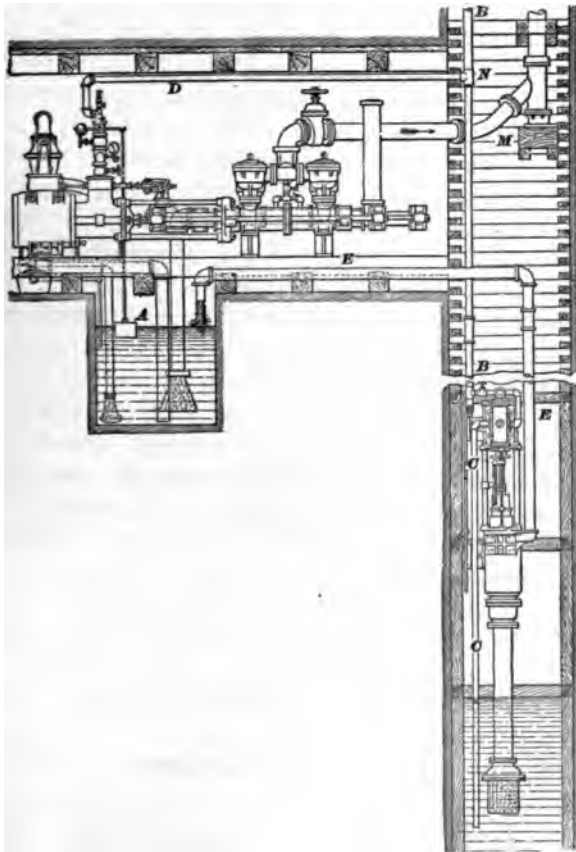
FIG. 789.

ch rises and lowers as the level of the water in *A* rises
. lowers.   This action operates a balanced throttle-valve
n the steam supply-pipe.   Should the water in the sump
ome too low, the float falls so far that the engine stops
omatically.   The movement of the float up or down also

governs the speed of the engine. K is an extra pump to be used in case it should be desirable or necessary to shut the other down. Its delivery-pipe J discharges into the same column-pipe O as shown. It is an excellent plan to have duplicate sets of pumps. There is then no need of stopping work should one of the pumps be disabled.

It will be noticed that the column-pipe is supported by a cast-iron stand, which rests upon the timbers M. This arrangement is shown more clearly in Fig. 789.

2262. Fig. 789 shows a pumping plant similar to the one described previously, except that only one pump is used, and a sinking-pump is employed to raise the water from the bottom of the shaft to the tank A. This arrangement is used when it is desired to sink a shaft below the level on which the pump stands, and the lift is too great for the sinking-pump to raise the water to the surface. As will be noticed, both pumps receive their steam from the same pipe B, which runs straight up the shaft to the surface. This pipe is covered with some material that is a non-conductor of heat, to reduce the loss of steam through condensation and to prevent the men from getting burned by accidentally touching it when working in the shaft. The exhaust steam pipe C of the sinking-pump discharges into the sump. E is the sinking-pump discharge-pipe. B is the steam-pipe; i supplies both the sinking-pump and the main pump by means of a **T** joint N and the pipe D. The manner of supporting the column-pipe is very clearly shown at M.

THE PULSOMETER.

2263. One of the most ingenious of the various machines operated by steam is the **pulsometer.** Fig. 790 shows a perspective view and Fig. 791 a sectional view of a pulsometer of the latest manufacture. In the sectional view the full lines represent the left-hand half, and the dotted lines indicate the position of the discharge-valves in the right-hand half of the pulsometer shown in Fig. 790. In the following description, the letters refer to both figures:

The steam-pipe is connected at E and the suction-pipe at S. C is an air-chamber, which has no connection with B and A, but communicates with the suction-pipe by means of the opening I, situated below the suction-valves F and G. The two latter valves are made of flat rubber, and are held to their seats, as shown in the cut, by means of the spindles R and T. The spindles are raised and lowered, as the case may require, by means of the nuts f and e. H, H are plates which may be removed to facilitate the examination of the valves. D is a hard-rubber ball, which acts as a valve for admitting the steam to the chambers A and B. M and N are exhaust-valves, also made of rubber, and situated in the chamber L, attached to the other half of the cylinder. They are raised and lowered in the same manner as the suction-valves, by turning the nuts g and h. K is the delivery or column pipe.

2264. The action of the pulsometer is as follows: Both chambers, A and B, are filled with water to about the height of the water in B, Fig. 791. The valve d is then opened, and the steam enters one of the two chambers A and B. Suppose it enters B, the valve D being at the right, as shown. The water in B will be forced through the delivery-valve N into and up the column-pipe K. This will continue until the water-level gets below the edge of the discharge-opening P. At this point the steam and water mix in the discharge-passage, and the steam is condensed, creating a vacuum in B. The pressure in A is now greater than in B, owing to the vacuum in B, and the ball-valve D is shifted to the left, the steam entering the chamber A, and driving the water through M into the passage O and column-pipe K in the manner just described. While this is being done, the pressure of the atmosphere forces the water up the suction-pipe S, opening the suction-valve F, and into the chamber B, filling it. When the suction-valve is closed, owing to the reshifting of the ball-valve D to the other side, the suction-water enters the air-chamber C through the inlet I, and is brought gradually to rest by the

compression of the air in C, thus preventing a shock, owing to the sudden stoppage of the inflowing water. When the water in A has reached the level shown, the steam in A is
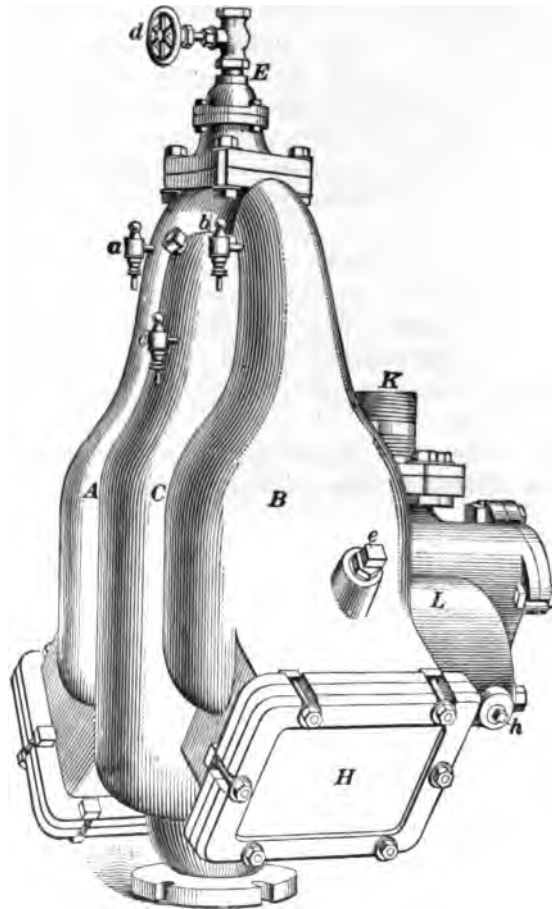
FIG. 790.

condensed, the ball D is shifted to the right, and B becomes the driving-chamber.

2265. In Fig. 790 are shown three small air-valves a, b, and c. The valve c admits air to the air-chamber C, to replenish that which is lost through leakage and through

absorption by the water. The valves *a* and *b* admit a small
quantity of air to the chambers *A* and *B*, respectively, just
before the suction begins. This injures the suction some-
what, but is necessary for two reasons : *First*, it acts as a
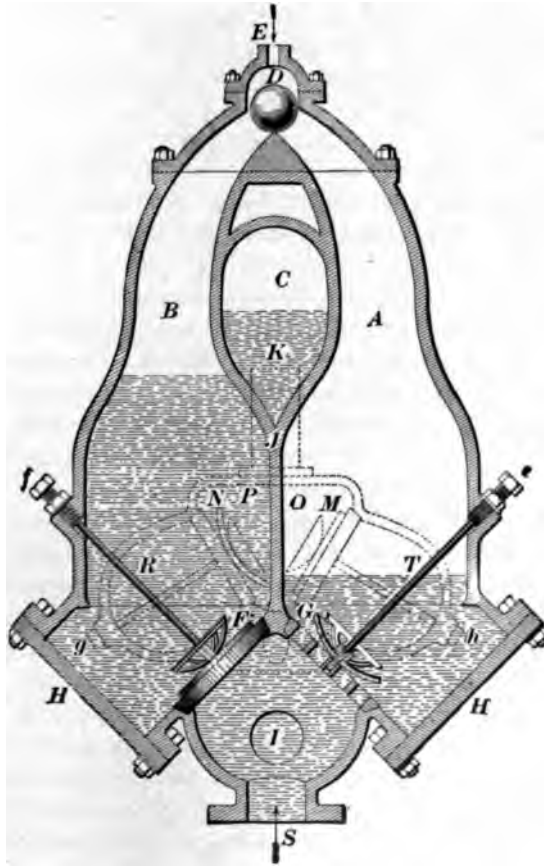
Fig. 791.

regulator, governing the amount of water admitted to the
chambers. *Second*, it prevents the steam from condensing
before the water gets below the edge of the discharge-outlet.
These valves open inwards, as before stated. Suppose there
is a vacuum in *A*, owing to the condensation of the steam.
The atmospheric pressure forces open the valve *a* and admits

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a little air to the cylinder.   The incoming water compresses this air, and soon closes the valve.   When the air has been compressed to such an extent as to balance the outside pressure of the atmosphere, the suction-valve G will close, and no more water can get in.   Since the same thing occurs in the other chamber, it is evident that the amount of air admitted controls the amount of water admitted during the suction period, more water entering when there is less air in the chamber, and *vice versa*.   The admission of the air is controlled by turning the valves a and b, and these can be so adjusted that the suction-valve in either chamber will close at the instant the ball is shifted to the other side, admitting the steam.

Moreover, the air prevents the steam from coming in contact with the water during the forcing process, until the water-level has sunk below the edge of the discharge-orifice. Air being a poor conductor of heat, the steam does not condense until the mixture of the steam and water has taken place.

2266.   The pulsometer will raise water by suction to a height of about 26 feet, although it is not advisable to exceed 20 feet, and force it, when necessary, to a height of 100 feet.   It has no wearing parts whatever except the valves, which are easily and cheaply repaired.   It will work in almost any position, and, when once started, requires no further attention.   There are no parts which can get out of order.   It will pump anything, including mud, gravel, etc., that can get past the valves.   Its first cost is low, and it requires no foundations to set up.   There is no exhaust-steam to make trouble, and no noise.   It uses more steam than a pump, its duty being from 7,000,000 to 10,000,000 foot-pounds per 100 pounds of coal.   One of the leading pump manufacturers of this country states that the *average* duty of single steams is from 15,000,000 to 20,000,000; of compound pumps about 30,000,000, and of compound condensing-pumps about 50,000,000 foot-pounds per 100 pounds of coal burned.

ELECTRIC PUMPS.

2267. All of the pumps heretofore described for underground mine work have been steam-pumps. The simple pump, both single and duplex, can be run by means of compressed air. As mentioned before, there are several ways by which the pumps may be driven other than by the use of steam. Of late, the electric pump is being used to some extent, and will, perhaps, in time, displace steam-pumps for underground mine use.

2268. A cut of an electric pump is shown in Fig. 792. It is what is termed a **triplex pump;** that is, there are three cylinders side by side, all three being operated at the same time from the same shaft A. B is the motor, the electric current being conveyed to it by two wires from a dynamo at the surface. As the motor revolves, it turns with it the shaft to which is keyed the pinion C. C gearing with D causes D to turn the pinion E, keyed to the same shaft as D. E gearing with F revolves the crank-shaft A, and with it the cranks G, H, and I, which impart a reciprocating motion to their plungers L, K, and J. These cranks are set at angles of 120° with each other, and the plunger-cylinders all discharge into the same delivery-pipe M, the consequence being that a nearly uniform discharge is secured, much better than that attained in the duplex construction, which is itself superior, in this respect, to the single pump. N is the suction-pipe. These pumps are made to raise water in single lifts from 400 to 800 feet, and to deliver at the point of discharge from 50 to 450 gallons per minute, according to size. In combination with these pumps, a small pump, called the **tail-pump,** is generally used to deliver the suction water to the main pump under a slight pressure, thus insuring the plunger-cylinder being full before the commencement of the return stroke.

The pump illustrated is termed a **horizontal triplex electric pump.** In many cases they are made vertical; that is, the plungers move vertically instead of horizontally. They are also made both triplex and duplex; the latter

type being applicable, with some modifications, to the ordi-
nary duplex steam-pump, the steam-cylinders being replaced
by the motor.

It is necessary, in order that the motor be effective, that
it revolve at a high speed, while the crank-shaft must turn
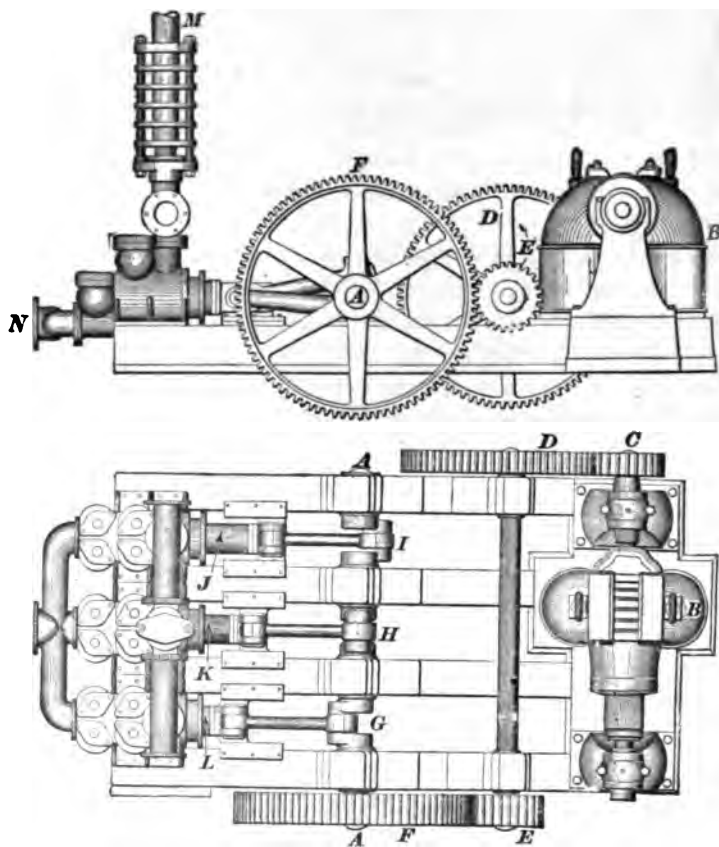
Fig. 792.

at a low speed; it is for this reason that the motion of the
motor is transmitted to the crank by means of gearing.
In Fig. 792 the gear D is about 3 times as large as the pin-
ion C, and F about $3\frac{1}{2}$ times as large as E; hence, the motor
revolves $3 \times 3\frac{1}{2} = 10\frac{1}{2}$ times while the gear F, and, conse-

quently, also the crank-shaft A, is revolving once. There-fore, if the crank-shaft A makes 50 revolutions per minute, the motor will make $50 \times 10\frac{1}{2} = 525$ revolutions per minute.

2269. Fig. 793 shows a **duplex electric sinking-pump.** E and F are the two plunger-rods, the plungers themselves being cen-tral-packed, as shown at H; D is the clamp-ing-piece for attach-ing the pump to the shaft-timbers, and G the eye-bolt for sus-pending it from a chain. A is the dis-charge-pipe; B, the suction-pipe, and C, the air-chamber. The only visible moving parts are short por-tions of the plungers and rods at H and E. No damage what-ever can come to this pump from the water. It will work just as well under water as out of it. The only objection on this score that can be urged against it is, that the wires which conduct the electricity to it may be broken by the falling debris. If proper care be exercised, this should not happen. They will raise water vertically 200 feet, and discharge from 100 to 300 gallons per minute, according to size.
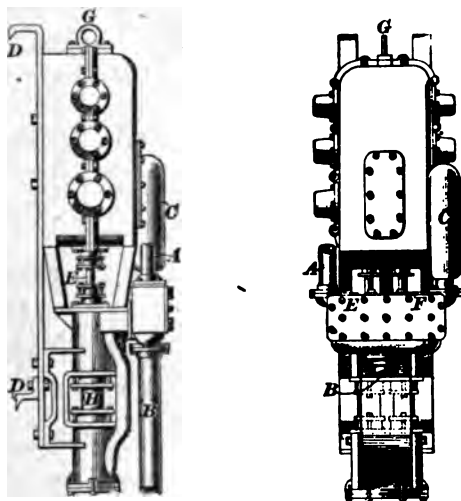
FIG. 793.

2270. A cut of a **water-power electric pumping-plant** is given in Fig. 794. A is a sinking-pump, which raises the water from the lowest level to the first station, dis-charging through the pipe C into the tank B. From this sta-tion, the water is raised to the next higher one by means of, in this case, **a vertical triplex pump,** and so on by one or more lifts to the surface. The wires which conduct the electricity down the shaft are enclosed in a small iron pipe
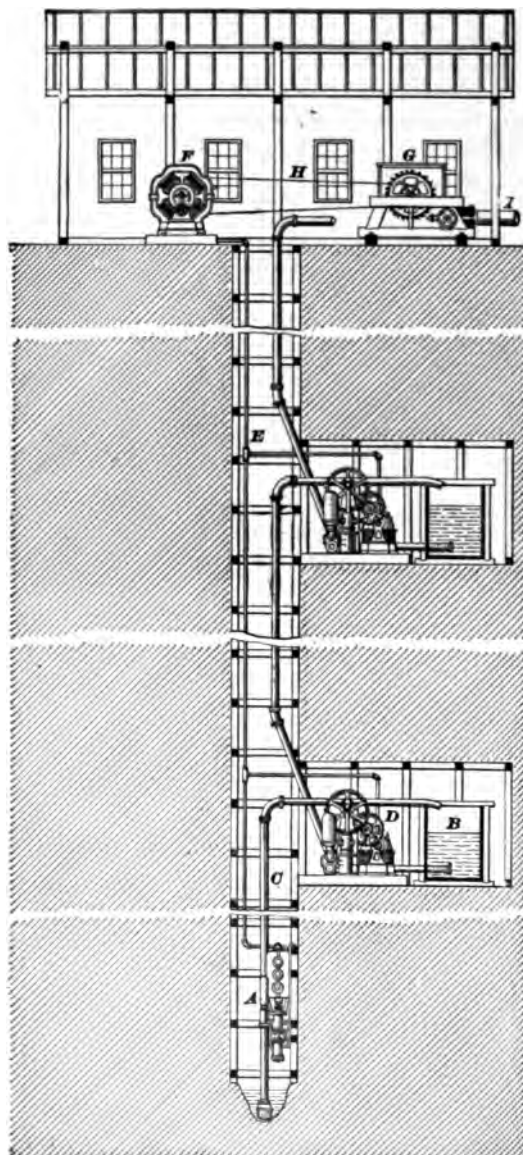
FIG. 794.

to prevent injury. *F* is the dynamo. *G* is a **Pelton ater-wheel,** the water being conducted to it through the ye *I*, and the power generated transmitted to the dynamo means of the belt *H*. It will, of course, be understood at a water-wheel can be used as a motive power only when natural head is available. It would not be advisable to put a plant of this kind for a head of less than, say, 40 or 50 et. In case a water-wheel can not be used, a steam-engine n be employed to drive the dynamo.

Instead of vertical triplex pumps, horizontal triplex or iplex pumps may be used; they take up more room than ie vertical type, and hence are not so convenient where ace is required.

HYDRAULIC PUMPING-ENGINES.

2271. It frequently happens in connection with mines iat water accumulates at a point below the level of the ater in the sump, and at a considerable distance from it, hich it is necessary to pump to the surface. The distance om the main pump may be half a mile or more, and the xpense of putting in a small pump, and conducting steam nd exhaust pipes to it, is out of all proportion to what it ould be under more advantageous conditions. In such ases as this, **hydraulic engines** are used. The principle nder which they operate is, that a small quantity of water lling from a great height will raise a larger quantity to a naller height.

2272. Before explaining the theory of the engine, the igine itself will be described, so as to make the theory sier to understand. Fig. 795 shows a hydraulic engine, e motor-cylinder *A* being shown in section. There are o valves, *B* and *C*, which are made of lignum-vitæ. *P* is e piston that drives the pump *E*, which may be of ordinary nstruction. *F* is an air-chamber on the discharge-pipe. is a flange for bolting the suction-pipe to the pump. *H* another flange for attaching the discharge-pipe. *T, T* are ors, which may be removed to allow examination of the

Fig. 706.
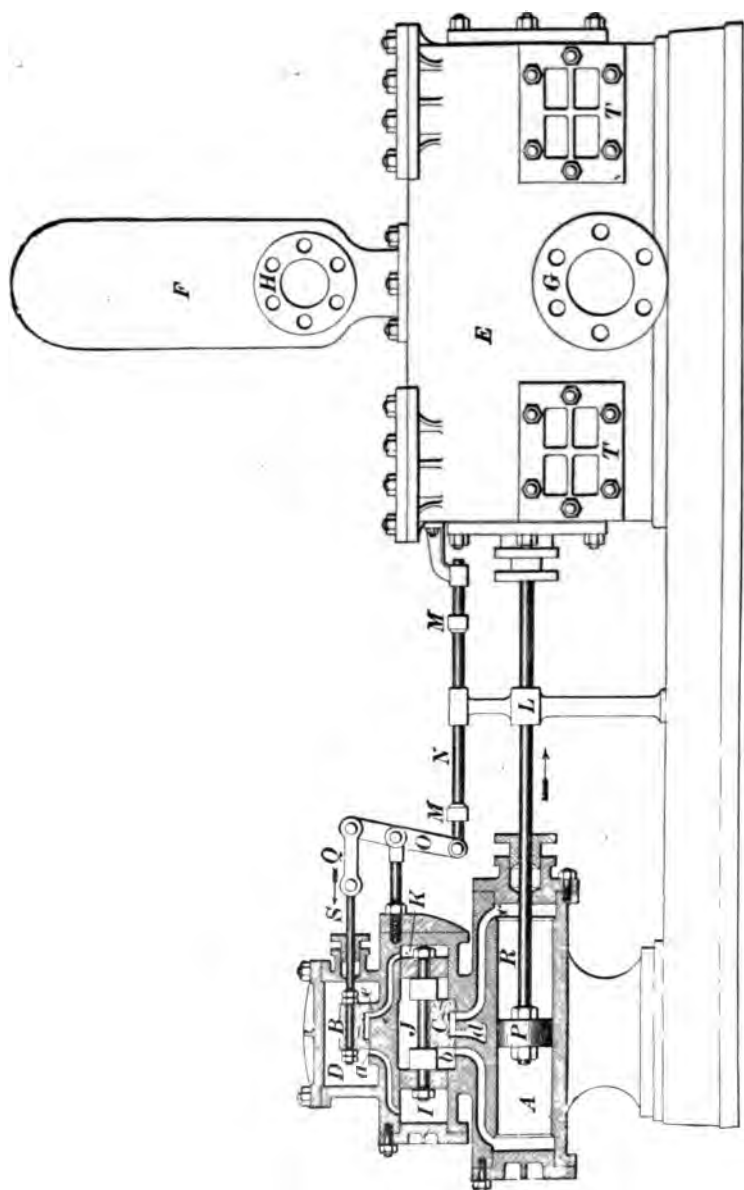
suction-valves. Suppose the piston, valves, etc., to be in
the position shown.   The water which drives the engine
fills the chest *D*, and has the full pressure due to its head;
it passes through the port *a* into the space *I*; there is also a
communication with the main valve-chamber *J* from which
the water passes through the port *b* into the cylinder, and
drives the piston in the direction of the arrow.   Attached to
the piston-rod is an arm *L*.   When the piston nears the end of
its stroke to the right, the arm *L* strikes the lug *M*, and causes
the rod *N* to actuate the lever *O*, one end of the lever being
attached to the rod *N*, and the other end to the valve-stem
S, by the link *Q*.   The valve *B*, pressed equally by the water
on both ends, is caused to be moved to the left, opening the
port *e*.   The water in the chest *D* then enters the space *K*,
and causes the valve *C* to be moved to the left, forcing the
water confined in the space *I* through the port *a* and the
under side of the valve *B* into the pipe which conducts away
from the pump the water discharged through the exhaust-
port *d*.   To better understand this last statement, suppose
the piston to be at the end of its stroke to the left, and that
the arm *L*, striking the lug *M*, has shifted the valve *B* to
the right, as shown in the cut.   The valve *C* will also move
to the right, for the reasons before given, and the water in
the space *K* will be forced through the port *e* and under the
valve into the water exhaust-pipe.

2273. In the case of hydraulic-engine pumps, the
motor-piston *P* is always smaller than the pump-piston, but
the length of stroke of both pistons is the same.   If there
were no friction or other resistance than that due to the
weight of the water, the areas of the two pistons would be
inversely proportional to the heads acting upon the pistons;
that is, if *a* be the area of the motor-piston (*P* in Fig. 795),
A the area of the pump-piston, *h* the head which acts upon
the pump-piston, and h_1 the head against which the pump
works (height of lift), the following proportion expresses the
relation between them:

$$a \; : \; A \; :: \; h_1 \; : \; h.$$

The amount which the pump will be required to discharge is usually known; also the heads h and h_1. The discharge being known, the length of the stroke and area of pump-piston can be so taken that the volume displaced by the piston in one stroke, multiplied by the number of required strokes per minute, shall be equal to the required discharge. When this has been done, the values of A, h_1, and h will be known, and a can be found from the proportion just given.

EXAMPLE.—Suppose that the head of water which acts upon the motor-piston is 640 feet, and the pump is required to discharge 80 gallons of water per minute under a head of 120 feet, what are the diameters of the motor and pump pistons, the length of their strokes, and the number of strokes per minute ?

SOLUTION.—Since one cubic foot of water contains 7.48 gallons, the number of cubic feet in 80 gallons would be $\dfrac{80}{7.48} = 10.695$ cu. ft. For a pump of this kind, it would be well not to have the piston speed exceed 80 feet per minute. Assume the number of strokes per minute to be 60, then the amount of water displaced in one stroke is $10.695 \div 60 = .17825$ cu. ft. $= .17825 \times 1,728 = 308$ cu. in. If the stroke be taken as 10 in. long, the area of the pump-piston will be $308 \div 10 = 30.8$ sq. in., and the diameter will be $\sqrt{\dfrac{30.8}{.7854}} = 6\frac{1}{4}$ in., nearly. Ans.

The piston speed will evidently be $\dfrac{60 \times 10}{12} = 50$ feet per minute. As this is well within the limit advised (80 feet per minute), it may be used, and, in case the pump should be required to deliver more than 80 gallons per minute at any time, the speed can be increased to meet the demand. To find the diameter of the motor-piston, first find the area by the proportion given above. In this case, $h = 640$, $h_1 = 120$, and $A = 30.8$; hence,

$$a : 30.8 :: 120 : 640, \text{ or } a = \frac{30.8 \times 120}{640} = 5.775 \text{ sq. in.}$$

The diameter, consequently, equals $\sqrt{\dfrac{5.775}{.7854}} = 2\frac{1}{4}$ in., nearly. Ans.

The values just calculated are theoretical values; the friction of the water in the pipes and the leakage past the piston will modify the results to a considerable extent; consequently, when calculating the sizes of a hydraulic pumping-engine, employ the method given in the latter part of this section, using formulas **190** to **196.**

In this arrangement, the water used in the motor-cylinder has to be raised again to the point where the pump discharges, and from there to the surface.

2274. Fig. 796 shows a hydraulic engine operated in a different manner, which is said to give excellent results, and may be worked as easily under water as out of it, should the mine be flooded. The motor-cylinder does not discharge the water remaining in the cylinder after the stroke is completed, into the sump, as described in the last figure, but uses the water over again, as the following description will show: *A* is a steam-engine, whose piston-rod *T* passes through the steam-cylinder *Q*, and also through two single-acting pump-cylinders *I* and *R*. The hydraulic engine is located at the bottom of the mine; it consists of two single-acting motor-cylinders *F* and *G*, and two single-acting pump-cylinders *H* and *S*. A small pipe *D* connects the pump-cylinder *I* with the motor-cylinder *G*, and another pipe *E*, of exactly the same size, connects the pump-cylinder *R* with the motor-cylinder *F*. *B* is a tank which contains sufficient water to charge the pipes *D* and *E*. *J* is the suction-pipe which conducts the water to the pump, and *K* is the column-pipe which delivers it to the surface.

2275. The action of the apparatus is as follows: Suppose that the pipes *D* and *E* are empty. The cock *U* is opened, and the water flows from the tank *B* until the pipes *D* and *E* are filled. The cock *U* is then closed, and the engine started. Let the engine be moving in the direction indicated by the arrow on the fly-wheel; the pistons in the cylinders *Q*, *I*, and *R* will then be moved towards the left. The piston in the cylinder *I* will force the water in *I* through the pipe *D* into the cylinder *G*. The piston in *G* is connected with the pistons in *S*, *H*, and *F* by means of the long piston-rod *W*. The pressure of the entering water against the piston in *G* forces it and the other three pistons in *S*, *H*, and *F* to the left. This action forces the water in the cylinder *H* through the discharge-valve *P* into the column-pipe *K*, causing it to discharge at *Y*. The vacuum

cheaper than in the case of direct-acting steam-pumps, and is said to give a higher efficiency than either compressed air or the electric wire.

Although the pumps may be used for lifts as high as 1,200 feet or more, the best results are obtained when the lift is about 600 feet.   The pressure in the pipe *D* and *E* is usually about 1,000 pounds per square inch.

2276.   Relative Merits of Underground Pumps.— The steam-pump is the most used of all the different classes of pumps, but, nevertheless, there are serious objections to its use.   In the first place, there is apt to be considerable loss due to the transmission of steam through long distances, as from the boilers at the surface to the pumps, perhaps half of a mile or more.   This is remedied to a great extent by covering the pipe with some non-heat-conducting material.   This increases the first cost quite materially, and renders it difficult to locate any leak that may occur without removing a considerable portion of the covering.   If the boilers are placed underground, near the pumps, the subsequent heat and gaseous products of combustion are a serious obstacle.   But the greatest objection is what to do with the exhaust-steam.   There are three ways of disposing of it: convey it to the surface through a pipe laid for that purpose; lead it into the sump, or discharge it into the upcast shaft.

2277.   When the pumps exhaust into the sump, it is often found that the whole body of water is heated to a comparatively high temperature, raising the temperature of the mine and increasing the humidity of the air to such an extent that the mine-timber decays with ruinous rapidity, and, at some collieries, the roof and coal on the airways, gangways, and travelingways are softened, and become both troublesome and dangerous.   A partial remedy for this would be to condense the steam, as shown in Fig. 788, before discharging into the sump.   Were this done, and the discharge-pipe located very near the pump suction-pipe, as

mentioned in connection with the figure, it is probable that the above-mentioned objections would disappear.

When the exhaust is conveyed to the surface through a pipe of considerable length, trouble is caused by the condensation of steam in the pipe, and also by the radiated heat. This condensation decreases the efficiency of the pump by increasing the back-pressure. The practice of conveying the exhaust into the upcast is often ruinous to the walls, roof, and timbering of the upcast passage.

In addition to the above, steam-pumps are useless when drowned out, either through breakage or sudden flooding. This risk can be remedied by having a surplus pumping capacity and reserve pumps in readiness.

All of the above objections can be eliminated by the use of compressed air in place of steam. Its efficiency is considerably less than that of steam, particularly when compound pumps are used.

2278. The use of electricity in connection with mine-pumps is so recent that it would be inadvisable to recommend it to replace any steam or compressed-air plant now in operation. In the near future, it is probable that it will supersede all other means of raising mine-water to the surface wherever new workings are being opened. The capability of the dynamo to be driven by both water-wheel and steam-engine, and the ease and efficiency with which the electric power can be transmitted to any point, as well as its comparatively low first cost, render this conclusion inevitable. Electric pumps will run equally as well under water as out of it—a great advantage in the case of a drowned-out mine.

SIPHONS.

2279. In many cases, water collects at some point in a mine higher than the level of the water in the sump or other place where it is desired to convey it. If there is an incline all the way, it may be conducted by means of a drain-pipe by gravity; but it frequently happens that, in order to

reach the sump, the water must be conveyed over a point higher than the level at the source, and it is not expedient to cut a passage through the high ground in order that the water may flow down by gravity.   In such cases, siphons may be used.

2280. The principle of the siphon is illustrated in Fig. 797.   Here A and B are two vessels, B being lower than A, and $A\ C\ B$ is the bent tube, or siphon.   Suppose this tube to be filled with water and placed in the vessels as shown, with the short branch $A\ C$ in the vessel A.   The water will flow from the vessel A into B as long as the level of the water in B is below the level of the water in A, and the level of the water in A is above the lower end of the tube $A\ C$.   The atmospheric pressure upon the surface of A and B tends to force the water up the tubes $A\ C$ and $B\ C$. When the siphon is filled with water, each of these pressures
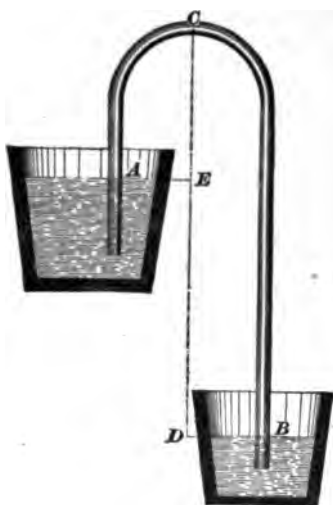
FIG. 797.

is counteracted in part by the pressure of the water in that branch of the siphon which is immersed in the water upon which the pressure is exerted.   The atmospheric pressure opposed to the weight of the longer column of water will, therefore, be more resisted than that opposed to the weight of the shorter column; consequently, the pressure exerted upon the shorter column will be greater than that upon the longer column, and this excess pressure will produce motion.

2281. In any siphon, the head which causes the flow of water is equal to the vertical distance between the two water-levels which the siphon connects; in the above figure, the head equals the distance $E\ D$.   Theoretically, the distance $C\ E$ of the highest point of the center of the pipe above the

level of the water into which the short leg of the siphon dips may be 34 feet; practically, 28 to 30 feet is the highest that a siphon will work successfully.   If required to work continuously, 21 feet should be the greatest height of $C\,E$.   The less this distance is, the better the siphon will work.   There is no limit to the distance $E\,D$ which constitutes the head.

2282.   Fig. 798 shows a siphon working in a mine.   It is desired to convey the water from D to E, the level of the water in E being always lower than in D.   The siphon consists of ordinary cast-iron pipe, jointed, and three valves, A, B, and C.   The suction end of the pipe is the same as the end
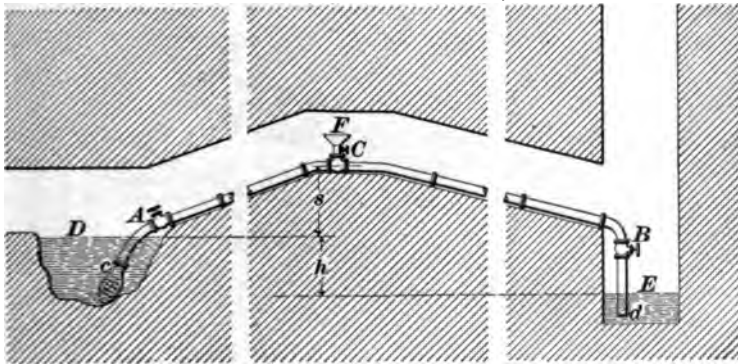
FIG. 798.

of the suction-pipe of a pump; i.e., it has a perforated pear-shaped end, in order to keep out large particles, which would prevent the siphon from working.   In order to start the siphon, it is necessary to remove the air in the pipe.   This is accomplished by closing the valves A and B, and opening the valve C.   Water is then conveyed to the funnel F and poured in.   The water drives the air out, and takes its place in the pipe.   When no more water can be poured in without overflowing at F, the valve C is closed, the valves A and B are opened, and the siphon is in operation.

The distance s between the highest point of the center of the pipe and the lowest level of the water in D must not exceed 28 feet; it would be better not to have it exceed 21 feet.

The greater the distance h between the two water-levels, the better the siphon will work.

Instead of filling the pipe with water in order to remove the air, an air-pump may be attached at C and the air pumped out. When this is done, and both ends of the siphon are submerged in water (as they should always be in practice), the valves A and B are left open, the water gradually rising in the pipe as the air is removed, and finally appearing at C; the valve C is at once closed, and the siphon begins to work.

2283. Fig. 799 shows a very convenient method of filling the pipe with water when the siphon discharges into the
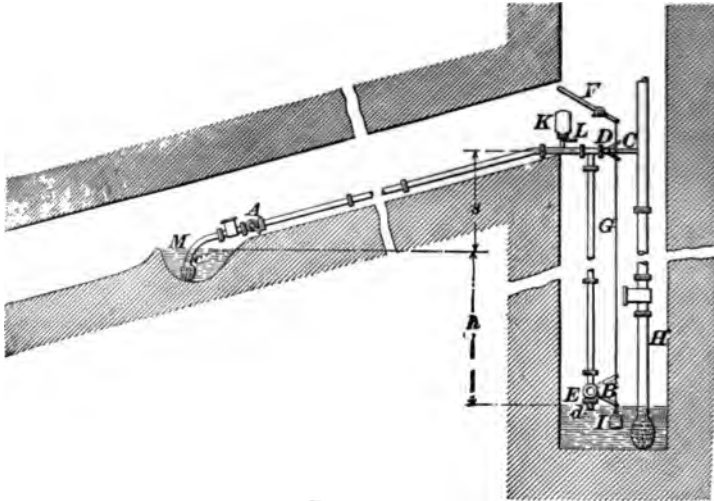
FIG. 799.

sump. Here M is the point from which it is desired to siphon the water, E is the sump, s is the height of suction, and h is the head which induces the flow. A is the valve at the suction-end, and B the valve at the delivery-end. H is the column-pipe of the main pump, C a small pipe leading from the column-pipe to the siphon, communication being opened or closed by aid of the valve D. K is a chamber to allow the air to escape, and L is a valve which controls the communi-

cation between the siphon and the chamber. All four valves, A, B, D, and L, are operated by handles, as shown. In order to keep the air from getting past the valve L when it is closed, thus entering the pipe and destroying the action of the siphon, the chamber K is kept filled with water.   The lever F has fastened to its short arm a rod G.   Attached to this rod are the handles of the valves B and D, and to its lower end the weight I.   When in the position shown, the valve B is open and D is closed.   In order to start the siphon, the valve A is closed and L is opened.   The lever F is then pulled down.   This action raises the handles of the valves D and B to the position shown by the dotted lines, opening the valve D and closing the valve B.   The water flows into the siphon from the column-pipe through the small pipe C.   When the siphon is filled, the water appears at chamber K, the valve L is closed, the lever F is released, and the weight I pulls it back into the position shown in the cut, closing the valve D and opening the valve B.   The valve A is then opened, and the siphon is in working condition.

2284.   In order that a siphon shall work properly, it is necessary that air should be kept out of the pipe, or, if it gets in, means should be provided for its escape.   Air will enter the
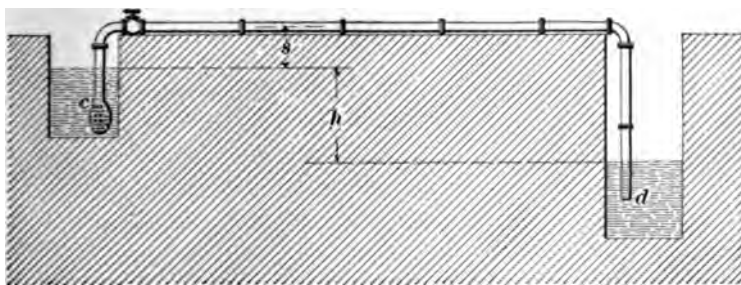
FIG. 800.

pipe in spite of all precautions, and, when once in, will collect at the highest point of the siphon because the pressure there is least.   The joints must be perfectly air-tight; even then the water absorbs air, which is given out again as the

pressure lessens. Then, too, the pipe seldom runs full continuously, and air enters the pipe with the water unless both ends of the pipe are submerged in water. Since the air always seeks the highest point of a siphon, sharp bends at this point, as at E, Fig. 801, should in all cases be avoided. A long bend, as in Fig. 798, a straight level pipe, as in Fig. 800, or a pipe on a long incline, as in Fig. 799, will always work well.

2285. A-siphon with a sharp bend, as in Fig. 801, will not work well, for the following reasons: As before stated, the air seeks the highest point, which is E. This air is at once compressed to an amount represented by difference
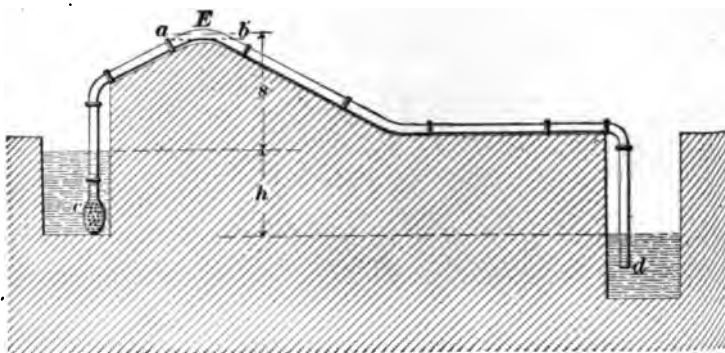
FIG. 801.

in pressures of a column of water whose height is s (in feet) and 34 feet (the height of a column of water which the atmosphere will support). Suppose that in the figure $s = 22$ feet; then the tension of the air at E would be 34 -- 22 = 12 feet of water = $12 \times .434 = 5.208$ pounds per square inch. This pressure will not be materially increased by the addition of a little more air, which, consequently, goes to increase the volume of the air already there. When the volume becomes sufficient to occupy the space $a E b$, the dotted line $a b$ representing a horizontal line just touching the bottom of the inside of the pipe, the water can not get through the bend, and the siphon is useless. The same is

o true of a siphon having a double bend, as shown in
g. 802.   Here the air will collect at E and F—at E first,
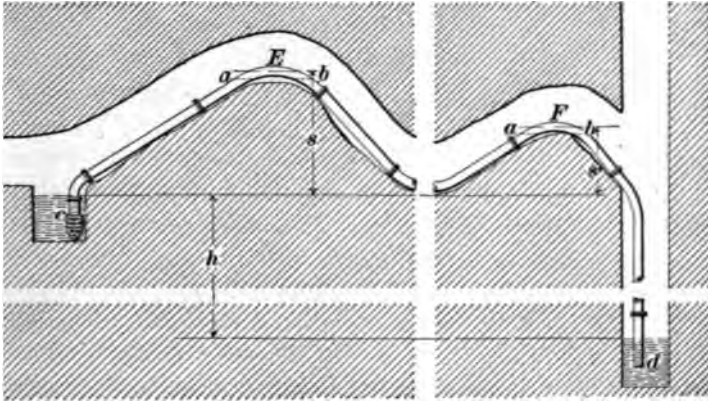
FIG. 802.

d F afterwards.   This is an extremely bad construction,
d should in all cases be avoided.

2286.   A device which will remedy the bad action of a
iphon to a considerable extent, by removing the air, is
own in Fig. 803.   Here A is an air-tight vessel connected
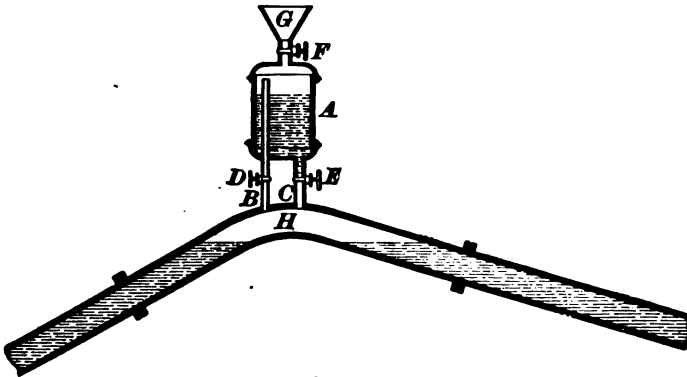
FIG. 803.

th the siphon by two pipes, B and C.   The pipe B extends
very nearly the top of A, while the pipe C barely enters
e bottom.   Each pipe has a valve, D and E.   On the top

of the vessel are a funnel G and valve F.   When the air has collected in the siphon and ceased its flow, the valves D and E are closed, and the valve F is opened.   Water is then poured into A until it is filled and overflows the funnel G. The valve F is closed and D and E opened.   Water will flow down through C, and the air will ascend through B, until the air is all out of the pipe.   This being done, D and E are shut and F opened.   The vessel is then filled with water, F is shut, D and E are opened and left open.   Any air which enters the siphon will, instead of collecting at H, seek the highest point by ascending B, and forcing out a certain amount of water through C.   This will continue until A is filled with air, when the valves D and E should be shut, and the vessel A refilled, as before described.   This arrangement may also be used to fill the siphon for the purpose of setting it to work.   It is, of course, evident that the highest point of the water in A must be not more than 28 feet above the level of the water at suction.

2287.   Theoretically, it makes no difference whether the discharge-end of a siphon is submerged or not, but practically it does, for the reason that, if the siphon is not flowing full, the air will enter and work its way to the highest point, unless the discharge-end is beneath the water.   It also makes no difference if one end of the siphon is larger than the main pipe.   On the contrary, it is rather an advantage to have the suction-end funnel-shaped, since the resistance encountered by the water on entering is thereby lessened.   A siphon will work better using cold water than when using warm water; hence, it works better in the winter than in the summer

2288.   The amount of water which a siphon will discharge is calculated by the formulas given for the discharge of a pipe.   The head is, in all cases, the distance marked h in Figs. 798 to 802, and the length l, used in the formulas, is the whole length of the siphon from the end of the suction-end to the end of the discharge-end.   In finding the head h, it is assumed that the discharge-end is submerged ; then the

head is the vertical distance in feet between the level of the water at suction and the level of the water at discharge.   If the discharge-end is not submerged, the head will be the vertical distance between the level of the water at the suction and the end of the discharge-pipe.   It makes no difference how far below the water the ends of the siphon may extend.   The two ends of the siphon may, in fact, be level. The head is measured as described, and the direction of the flow will always be from the higher to the lower *water-level.* The length of the pipe is, in all cases, measured from *c* around to *d.*

EXAMPLE.—A siphon has a total length of 1,420 feet; its diameter is 4 inches, and the distance between the water-levels is 38 feet.   What is the discharge in gallons per hour ?

SOLUTION.—It is first necessary to find the velocity by formula **182.**

Thus, $v_m = 2.315 \sqrt{\dfrac{h\,d}{f\,l}} = 2.315 \sqrt{\dfrac{38 \times 4}{.025 \times 1,420}} = 4.79$ ft. per sec.

From Table 45, $f = .023$ for $v_m = 4$, and $.0214$ for $v_m = 6$; difference $= .023 - .0214 = .0016$ for a difference of 2 feet per second in the velocity $= .0016 \div 2 = .0008$ for a difference of 1 foot per second in the velocity. $4.79 - 4 = .79$. $.0008 \times .79 = .000632 =$ amount to be subtracted from .023 to give the value of f for $v_m = 4.79$.   Hence, $f = .023 - .0006 = .0224$, using but four decimal places.   Substituting the values of f, l, h, and d in formula **186,**

$$Q = .09445\, d^2 \sqrt{\frac{h\,d}{f\,l + .125\,d}} = .09445 \times 4^2 \sqrt{\frac{38 \times 4}{.0224 \times 1,420 + .125 \times 4}} =$$

3.2778 gal. per sec.' $= 3.2778 \times 60 \times 60 = 11,800$ gal. per hour, very nearly.   Ans.

CALCULATIONS PERTAINING TO PUMPS.

2289.   To find the pressure in pounds per square inch corresponding to any given head of water:

Rule.—*Multiply the head in feet by .434; the result is the pressure in pounds per square inch.*

2290.   To find the head of water corresponding to a given pressure in pounds per square inch:

Rule.—*Multiply the given pressure in pounds per square inch by 2.304; the result is the head in feet.*

EXAMPLE.—What pressure will a head of 120 feet of water exert?

SOLUTION.—Applying the first rule,

$$120 \times .434 = 52.08 \text{ lb. per sq. in.}\quad \text{Ans.}$$

EXAMPLE.—What head of water will exert a pressure of 65 lb. per sq. in.?

SOLUTION.—Applying the second rule,

$$65 \times 2.304 = 149.76 \text{ feet.}\quad \text{Ans.}$$

2291. To find the size of the plunger-cylinder to discharge a given number of gallons per minute:

Let G = number of gallons discharged per minute;
  S = plunger speed in feet per minute;
  d = diameter of cylinder in inches.

Then, $$d = 4.95 \sqrt{\frac{G}{S}}.$$

Since, however, there is always more or less slip of the water past the plungers, it is usual to add ⅕ of the required number of gallons to the value given to G in the above formula, to allow for this slip. Doing so, the formula becomes

$$d = 5.535 \sqrt{\frac{G}{S}}. \qquad \textbf{(190.)}$$

Formula **190** should always be used when calculating the size of the plunger-cylinder to discharge a certain number of gallons per minute. The piston speed is the number of feet traveled per minute by the plunger when forcing water; that is, it equals the length of the stroke in feet multiplied by the number of working strokes per minute. If the pump is double-acting, the number of working strokes is the same as the total number of plunger-strokes, both forward and back; if single-acting, half of that number.

EXAMPLE.—What should be the diameter of a pump-plunger required to discharge 130 gallons of water per minute, the speed of the plunger to be 115 feet per minute? If the pump is double-acting and the stroke is two times the diameter, how many strokes must it make per minute, and what is the length of the stroke?

SOLUTION.—Applying formula **190**,

$$d = 5.535 \sqrt{\frac{G}{S}} = 5.535 \sqrt{\frac{130}{115}} = 5.88 \text{ in., say, } 5\tfrac{7}{8} \text{ in.} \quad \cdot \text{Ans.}$$

Since the stroke is twice the diameter,

$$\text{stroke} = 5\tfrac{7}{8} \times 2 = 11\tfrac{3}{4} \text{ in.} = \frac{11.75}{12} \text{ ft.} \quad \text{Ans.}$$

$$\text{Number of strokes} = 115 \div \frac{11.75}{12} = \frac{115 \times 12}{11.75} =$$

117.44 strokes per minute, nearly.   Ans.

This speed is rather high for a pump, and should be em-
ployed only when absolutely necessary.

2292. To find the approximate discharge in gallons
per minute of a mine-pump, when the diameter and plunger
speed are known, use the following formula:

$$G = .03264 \, d^2 \, S. \qquad \textbf{(191.)}$$

The same allowance has been made for the slip in this
formula that was made in formula **190.** If the theoretic
discharge is required, $G = .0408 \, d^2 \, S.$

EXAMPLE.—What is the probable discharge of a duplex double-
acting mine-pump whose plungers are 10 inches in diameter, stroke
24 inches, and which makes 40 strokes per minute ?

SOLUTION.—Applying formula **191,**

$G = .03264 \, d^2 \, S = .03264 \times 10^2 \times (2 \times 40) = 261.12$ gal. per min.,
since 24 in. = 2 ft. and the piston speed = 2×40 ft. per min. The total
discharge is twice this amount, or

$$261.12 \times 2 = 522.24 \text{ gal. per min.} \quad \text{Ans.}$$

EXAMPLE.—In the above example, what is the theoretic discharge ?

SOLUTION.— $G = .0408 \, d^2 \, S = .0408 \times 10^2 \times (2 \times 40) = 326.4$ gal. per
min. $326.4 \times 2 = 652.8$ gal. per min. Ans.

2293. To find the horsepower of a steam or air cylinder
to discharge a certain number of gallons of water per minute
with a given lift, substitute in the following formula, in
which $H =$ the number of horsepower, $h =$ vertical height
in feet between the highest point of the center of the deliv-
ery or column pipe and the level of the water in the sump
or place from which it was taken, and $G =$ the number of
gallons discharged per minute :

$$H = .00038 \, G \, h. \qquad \textbf{(192.)}$$

The theoretic horsepower will be two-thirds of the above result

EXAMPLE.—How many horsepower should the steam-cylinder of a pump be designed for which is required to discharge 350 gallons per minute, the total lift being 320 feet ?

SOLUTION.—Applying formula **192,**

$H = .00038\ G\ h = .00038 \times 350 \times 320 = 42.56$ horsepower.   Ans.

In the above example, the theoretical horsepower is $42.56 \times$ $\frac{2}{3} = 28.37$ horsepower.   In formula **192,** allowance has been made for friction of water in the pipe, engine friction, pump friction, etc.

2294. If it is desired to know the height which a pump will raise water, when the horsepower of the steam-cylinder and discharge of the pump have been determined, use the following formula, in which the letters have the same meaning as in formula **192:**

$$h = \frac{H}{.00038\ G}. \qquad (193.)$$

EXAMPLE.—To what height will a 40-horsepower pump force 280 gallons of water per minute ?

SOLUTION.—Applying formula **193,**

$$h = \frac{H}{.00038\ G} = \frac{40}{.00038 \times 280} = 376 \text{ feet.}   \text{Ans.}$$

2295. To find the size of the steam or air cylinder of a pump, first calculate the horsepower by formula **192,** then proceed as follows: It is customary to design pumps on a basis of 100 feet plunger speed per minute.   For mine-pumps working continuously, this is about right, although in some cases as high as 240 feet per minute has been attained.   Nevertheless, for continuous working, 100 feet per minute is a fair allowance, and does not bring excessive strains on the pump.   If a simple pump is to be used, the mean pressure of the steam or air will be the same as the gauge pressure at the pump, since the pressure is carried full stroke.   If the pump is also direct-acting, the speed of the steam-piston will be the same as the pump-piston or plunger speed.

Let S = piston speed;
    D = diameter of cylinder in inches;
    r = ratio between the length of stroke and diameter of cylinder;
    l = length of stroke in feet;
    N = number of strokes per minute;
    H = horsepower;
    P = steam or air pressure per sq. in.

Then, $$D = 205 \sqrt{\frac{H}{PS}}. \qquad (194.)$$

The diameter may also be found by formula **148**, *Air and Air Compression*, Art. **2152**.

$$D = 79.6 \sqrt[3]{\frac{H}{rPN}}.$$

Having obtained the diameter by means of either formula **148** or formula **194,** the stroke can be found by multiplying the diameter by the value of the ratio r.   In case formula **194** is used, the number of strokes can be found by dividing the piston speed by the length of the stroke in feet.

EXAMPLE.—A pump to be driven by compressed air, at a pressure of 45 pounds per square inch, is to have a piston speed of 100 feet per minute.   If 32 horsepower are required to operate it, what should be the size of the air-cylinder and the number of strokes?

SOLUTION.—Using formula **194,**

$$D = 205 \sqrt{\frac{32}{45 \times 100}} = \sqrt{\frac{42{,}016.8 \times 32}{45 \times 100}} = 17.285 \text{ in., or say } 17\tfrac{1}{4} \text{ in.}$$

For this case, let the stroke be, say, 22 in., thus making r a little over 1¼.   The number of strokes will then be $100 \div \tfrac{22}{12} = 54\tfrac{6}{11}$.   To make even figures all around, let the number of strokes be 55 per minute.   Ans.

Since it is easier to extract square than cube root, the first of the above formulas is to be preferred to the second.

SIZES OF SUCTION AND DELIVERY PIPES.

2296.   The usual practice is to allow a velocity of 200 feet per minute in the suction-pipe, and 400 feet per minute in the delivery-pipe.   Substituting these values for S in the formula for the theoretical diameter of the plunger given in

Art. **2291,** and letting d_1 be the diameter of the suction pipe and d_2 the diameter of the delivery-pipe,

$$d_1 = 4.95 \sqrt{\frac{G}{200}}, \text{ or } d_1 = .35 \sqrt{G}. \qquad \textbf{(195.)}$$

$$d_2 = 4.95 \sqrt{\frac{G}{400}}, \text{ or } d_2 = .25 \sqrt{G}. \qquad \textbf{(196.)}$$

. The pipes may be larger than the values calculated by the above formulas, particularly the suction-pipe, but it is not a good plan to make them any smaller. The larger the pipes are, the less the velocity, and, consequently, the less the frictional resistances.

EXAMPLE.—What should be the diameters of the suction and delivery pipes of a pump which discharges 225 gallons of water per minute?

SOLUTION.—Applying formula **195,**

$$d_1 = .35 \sqrt{G} = .35 \sqrt{225} = 5.25 \text{ in.}$$

Since the pipe sizes usually vary by even inches above 4 in. diameter, the size of the suction-pipe should be either 5 in. or 6 in. diameter, preferably the latter, but assumed for this case to be 5 in. Also $d_2 = .25 \sqrt{225} = 3.75$ in., or say 4 in. diameter. Hence, the diameters are, respectively, 5 in. and 4 in. **Ans.**

Some manufacturers make the diameters of their suction-pipes equal to the diameter of the plunger.

EXAMPLE.—Calculate the sizes of the steam and water cylinders and of the suction and delivery pipes for a direct-acting steam-pump to discharge 770 gallons of water per minute, under a head of 1,000 feet, single lift, the steam-pressure at the pump to be 71 pounds per square inch.

SOLUTION.—Assume for this case a plunger speed of 120 feet per minute. Then, using formula **190,**

$$d = 5.535 \sqrt{\frac{G}{S}} = 5.535 \sqrt{\frac{770}{120}} = 14.05 \text{ in., say 14 in., } =$$

diameter of plunger.

Assuming the stroke to be three times the diameter of the plunger, the stroke $= 14 \times 3 = 42$ in. Since the pump is direct-acting, the stroke of the steam-piston must also be 42 in. The number of strokes per minute will be $120 \div 4\frac{2}{3} = 34\frac{2}{7}$. The horsepower is found by means of formula **192.**

$$H = .00038 \, G \, h = .00038 \times 770 \times 1,000 = 292.6 \text{ horsepower.}$$

iameter of the steam-cylinder is found by formula **194.**

$$D = 205\sqrt{\frac{292.6}{71 \times 120}} = 38 \text{ in., very nearly.}$$

eter of suction-pipe, $d_1 = .35\sqrt{G} = .35\sqrt{770} = 9.712$ in., say

eter of delivery-pipe. $d_2 = .25\sqrt{770} = 6.937$ in., say 7 in.

,
Diameter of plunger,	14 in.	
Diameter of steam-cylinder,	38 in.	
Diameter of suction-pipe,	10 in.	
Diameter of discharge-pipe,	7 in.	Ans.
Stroke of pump,	42 in.	
Number of strokes per minute,	34¾.	
Horsepower,	292.6.	

PLE.—Find the sizes of a duplex pump to fulfil the same
ns given in the last example, assuming the stroke to be about
the diameter of the plunger, and the steam-pressure to be
ds per square inch.

ɪoɴ.—The total horsepower and the diameter of the discharge-
mmon to both) will be the same as before. Each pump-
will discharge $\frac{770}{2} = 385$ gallons per minute.

e, applying formula **190,**

$$d = 5.535\sqrt{\frac{385}{120}} = 10 \text{ in., very nearly.}$$

e $= 10 \times 2\frac{1}{2} = 25$ in., say 24 in., or 2 ft.
ɪumber of strokes per minute $= 120 \div 2 = 60.$

power of each steam-cylinder $= \dfrac{292.6}{2} = 146.3.$

ɪing formula **194,**

$$05\sqrt{\frac{146.3}{76 \times 120}} = 25.9 \text{ in., the diameter of the steam-cylinder.}$$

ɪ.

uction-pipes each deliver 385 gallons per minute to the pump;
pplying formula **195,**

$$d_1 = .35\sqrt{385} = 6.87 \text{ in., say 7 in.}$$

quently,
Diameter of plungers (2),	10 in.	
Diameter of steam-cylinders (2),	26 in.	
Diameter of suction-pipes (2),	7 in.	
Diameter of discharge-pipe (1),	7 in.	
Stroke of pump,	24 in.	Ans.
Number of strokes per minute,	60.	
Horsepower of each cylinder,	146.3.	
Total horsepower,	292.6.	

2297. Starting a Pump.—See that the pump is well oiled, and that all pipes and connections are free from obstructions; see that the stuffing-boxes and plungers are properly packed. Open the charging and relief pipes, to fill the suction and cylinders with water and drive out the air; also, open the drain-cocks of the steam-cylinder. Before starting, allow the steam-cylinder to warm up thoroughly. Turn on the steam gradually, and run the pump at slow speed for a short time.

Failure of a pump may be due to a multiplicity of causes, the chief of which are the following: Air in the pump-chamber or in the suction-pipe; dirt in the suction-pipe or in the valves; suction-pipe too long or too small and crooked ; air-chamber too small; leaky steam-valves or warm pistons or plunger; pump hot and filled with vapor; improper design of pump for the work to be done.

The greatest difficulty met with in pumping mine-water is the chemical action of the water upon the pump-cylinders, plungers, rods, valves, etc. It is particularly destructive in the anthracite coal regions. Portions of old mine-pumps are seen in which some of the cast-iron parts were so soft that they could be easily cut with a knife. They have the appearance of a honeycomb. Other instances have been frequently noted where the water has dripped on a steel rail from the roof of a mine, and eaten a hole through it. When the mine-water is in this condition, the life of a pump is short at the best. The exposed cast-iron parts are made of the hardest cast iron that can be worked. The water will attack wrought iron and steel even quicker than cast iron. This chemical action is much less rapid on gun-metal, phosphor-bronze, and several other alloys; hence, in well-constructed pumps for raising acid water of this kind, the valves are made of this material when not made of rubber. In cases of this kind, and also when the water is very gritty, the cylinders are bored larger than the plunger, and a gun-metal or phosphor-bronze shell, about an inch thick, is inserted. The wear comes principally on the bottom, and when it has worn more than desired there, the shell can be turned partly around, so that the wear may come at another point. When worn out, the plunger and shell can be replaced, and the old shell melted up.

MINE HAULAGE.

HAULAGE SYSTEMS.

INTRODUCTION.

2298. Underground haulage, whether done by wire rope or otherwise, is always carried on in two distinct stages. The first or local haulage is done by drawing the cars from the working face to a gathering-up or central station, from which the general haulage begins. From the latter station the loaded cars are hauled in trains to the bottom of the shaft or slope, or out of the drift, as the case may be. To secure economy and despatch, it is necessary that the local haulage be made as short as possible, as this work is generally done by mules, and is more costly than mechanical haulage. For the same reason, the general or mechanical haulage is made as long as possible.

2299. This section deals principally with wire-rope haulage. There are four classes of wire-rope haulage which will constitute the principal divisions of the discussion. They are :

1. Gravity-planes;
2. Engine-planes;
3. Tail-rope systems; ·
4. Endless-rope systems.

GRAVITY-PLANES.

CONDITIONS REQUIRED FOR SUCCESSFUL OPERATION.

2300. This system of haulage is done by gravitation, and, as far as the motive power is concerned, it might be supposed that it is cheap and economical, but such a conclusion is not always the correct one, for very important reasons, which should be known.

§ 22

A prominent railway company for some years did its coal haulage on the surface by a series of gravity-planes or self-acting inclines. In course of time, it was found that the work between the terminals could be more cheaply done on a properly graded road by steam-locomotive haulage than by gravitation. The reason for this was that many of the incline roads were short, and the number of persons employed on each short incline made in the aggregate a great number ; the repeated stoppages for detaching one set of ropes and attaching another entailed a considerable waste of time, and it was found impossible to keep all the inclines running in such accord that the train from one would arrive in time to follow that of another. The result was that the quantity of coal hauled per day was relatively small, being only about one-fourth of what could be hauled by steam-locomotive haulage. It was also found that the cost of ropes, rollers, and the services of the men employed far more than counter-balanced the cost of fuel and other expenses incidental to locomotive haulage. These same conditions occur in the mine, and in the same way the limitations of a costless power sometimes cause stoppages that reduce the output to such an extent that either direct steam-power or transmitted power is found to be better and cheaper. There are conditions under which gravity-planes are cheap and effective, but these are seldom found in the principal or primary haulage, excepting when the self-acting incline haulage is done with an endless rope.

2301. The haulage on self-acting inclines where the pitch is heavy is done with a pair of ropes and a pair of drums, by which arrangement the trains can be kept under perfect control with the brake, as no slipping of the rope on the drums can occur. Where the pitch is light, a single rope is used, in which case the rope is given one turn upon the head-wheel. This is found to be quite sufficient, for under such conditions it is not necessary for the brake to be so tightly applied as to cause the one coil of rope to slip.

2302. Until quite recently, the only self-acting inclines in use were those just noticed, but now self-acting inclines with endless ropes are fast displacing them. Each of the two rope systems is subject to very sharply defined limits, because the rope reaching to the bottom of the incline soon weighs as much as the descending coal, and then the gravity of the coal ceases to supply the required motive power.

2303. For example, if on an incline with 4 per cent. grade, the rope reaching to the foot weighs 2,000 lb., a loaded car 4,000 lb., and an empty car 1,500 lb., the loaded car will not exert force enough to pull the empty car up, for the following reasons :

First, the friction, which amounts to about $\frac{1}{40}$ of the load, must be considered; second, the fact that the descending car balances the ascending car must be borne in mind; therefore, the force is exerted only by the coal in the loaded car.

The resistance offered by the rope is caused by (a) its weight, and (b) the friction due to its weight. To move the rope up the incline regardless of friction requires $2,000 \times .04$, or 80 lb. To this must be added the friction, which amounts to $\frac{2,000}{40}$, or 50 lb., making the total force required to move the rope equal to $80 + 50 = 130$ lb.

Now, the force required to move the rope must all come from the weight of the coal in the loaded car. On a 4 per cent. incline the 2,500 lb. would exert a force parallel to the incline equal to 4 per cent. of 2,500 lb., or 100 lb. From this must be subtracted the friction due to both the loaded and empty cars, or $\frac{4,000 + 1,500}{40} = 137.5$ lb. Now, we know that we can not subtract 137.5 from 100, and, therefore, it is evident that the loaded car is entirely too light to start the empty car and rope from the bottom.

To make the matter more clear, let the grade be 6 per cent., and the weights of rope, cars, and coal be the same as in the previous example. Now, the force required to move

the rope will be equal to $(2,000 \times .06) + \dfrac{2,000}{40} = 170$ lb

The pull exerted by the 2,500 lb. of coal will be $2,500 \times$ $.06 = 150$ lb. The friction due to the coal and the two cars will be $\dfrac{4,000 + 1,500}{40} = 137.5$ lb. This must be subtracted from the 150 lb. to obtain the net force due to the weight of the coal; thus, $150 - 137.5 = 12.5$ lb. Since the coal in one loaded car, under the conditions given, exerts a working force of but 12.5 lb., it is plain that it will be necessary to run several cars in a trip to get force enough to overcome the friction of the rope. As the force necessary to move the rope is 170 lb., it will require in each trip $\dfrac{170}{12.5} = 13.6$, or 14 cars.

2304. When gravity-planes are run with a pair of ropes, the grade should increase as the length increases. This increase, however, can not always be secured, because we must take a grade as we find it. The length of an incline may be increased until the number of cars in the train can not lift the heavy rope. This conclusion is apparent when it is understood that if the weight of the rope per foot remains the same, and if the length of the incline is double, the number of cars in a train must be doubled. This statement, however, still falls short of the exact truth, for as the number of cars in a train increases in number, the weight per foot of the rope for such trains must also increase, and the result is that gravity-planes exceeding half a mile in length are seldom found, except where the pitch is 30° or more.

It is now clear that for an incline to be self-acting the useful gravity force (or the force that remains after the friction due to the weights of a double train of cars and the load in one of them has been subtracted from the weight that gravitates) must exceed the gravity weight of the rope and the friction due to its weight.

2305. There are cases in the local or secondary haulage of a mine where a gravity-plane is of great value. For

instance, where the working face is advancing up grade, self-acting inclines called "jigs" are adopted. These are self-acting inclines in which a balance-weight is pulled up by the descending full car, and the empty car in turn is pulled up to the working face by the balance-weight, or jig-weight, as it is generally called. In some of these, the loaded truck or jig runs on narrow-gauge rails between the rails of the ordinary track, and in other cases the jig is made to run on a track in a parallel opening. Short, self-acting inclines are also used to advantage in running loaded cars from a counter gangway to the main gangway, driven at a lower level.

DESCRIPTION OF DETAILS.

2306. Fig. 811 shows an ordinary self-acting incline, or one in which the weight of the coal in a loaded train acts as a motive force. At the head of the incline is seen the drum, or reel A, around which the ropes wind. When a single drum of this character is used, one of the ropes runs off the top side, and the other runs off the bottom side of the drum; a moment's consideration will explain the reason for this. If both ropes were on one side of the drum, they would both run off or on together, but as they coil on opposite sides of the drum, one runs on while the other runs off.

2307. The incline shown in Fig. 811 is provided with double tracks, to allow the empty trains to pass each other without danger of colliding. Between the rails are seen the rope-rollers I, I, used to prevent the ropes trailing on the ground. Trains on a self-acting incline begin to accelerate in speed at a point midway in the plane, and after the loaded cars have reached the lower side, instead of the weight of the rope reacting against the gravity force of the coal, it now supplements that force, and the result is an increased speed. Unless something is done to provide for checking the acceleration of the cars, dangers of a manifold
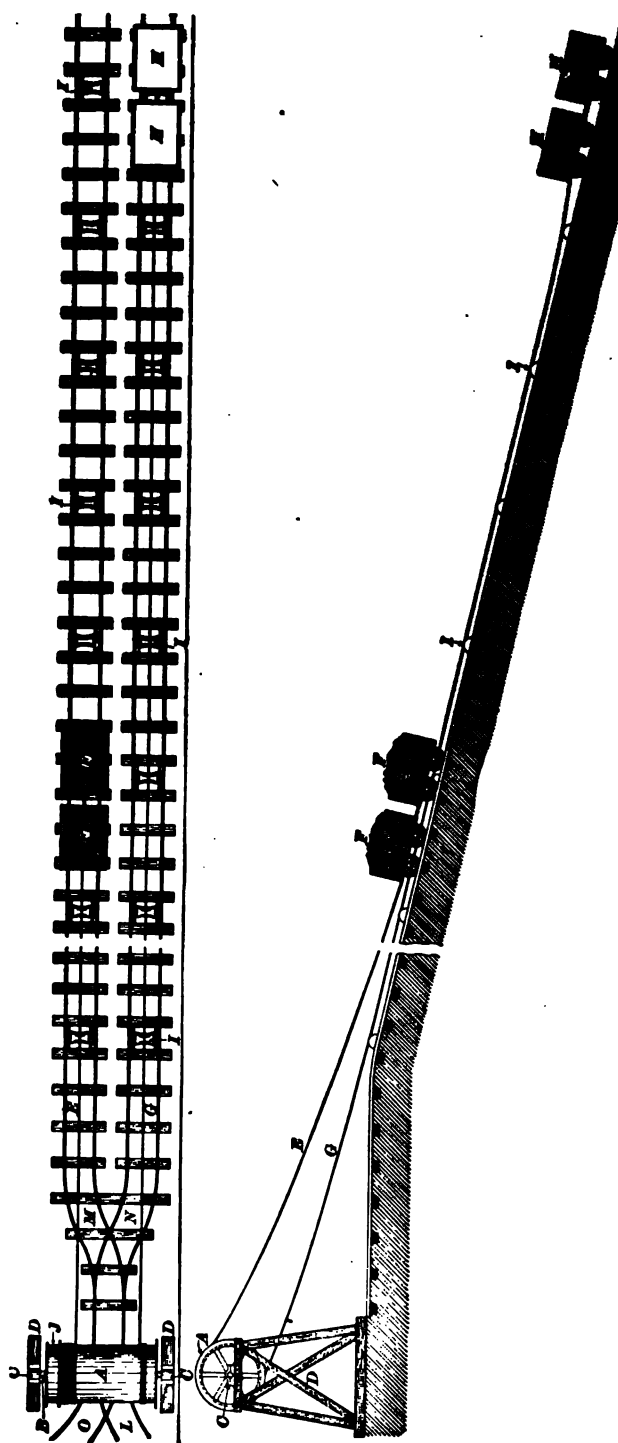
haracter may occur.   For this reason, a brake *J* is provided,
·hich is applied by an attendant at the drum.   For running
ιe full cars onto the incline and the empty ones off of it,
·oper branch tracks and switches are necessary.   For ex-
aple, an automatic switch is placed at the head of the
uble tracks in such a way that the loaded cars, on reach-
ʒ the top of the incline, may alternately take the tracks
 and *N*.   When the empty train *H H* reaches the head of
ᴇ incline passing out of the track *N*, it passes on to the
ιpty car line *O*; then two loaded cars passing out of *L* are
tomatically switched onto *N*, for the empty cars, in pass-
ʒ out of *N*, set the switch for the loaded cars to run onto *N*.

2308.   A vertical section of the incline is seen in the
ᴡer portion of Fig. 811.   The loaded cars are descending
 F F, and the empty cars are ascending at *H H*.   The
pe from the top of the reel is attached to the full cars *F, F*,
ιd the rope from the under side of the reel is attached to
 e empty cars *H, H*; consequently, the drum, as seen in
 e end view, is turning in the direction of the hands of a
atch; that is, running the rope off the top and running it
ι at the bottom.   The head-frame for carrying the drums
such as may sometimes be seen on the surface, and also
the mine.   Grip-wheels and fleet-wheels have in a great
 any cases displaced the head-frame, but there are cases in
ick seams where it is still used with advantage.

2309.   Fig. 812 shows an under level grip or fleet
ιeel, situated under a surface structure.   In principle it is
ᴇ same as the fixings of a grip-wheel under the level of the
ιcks at the head of an incline in the mine, for there an un-
rground chamber is cut out for the location of the wheel,
shown in Fig. 820.   Fig. 812 also illustrates an arrange-
ᴇnt of the tracks differing from that shown in Fig. 811.
 this case a treble rail is continued to the parting, and
ιm the lower end of the parting to the foot of the incline.
 the majority of underground inclines, the arrangement
 the tracks is like that illustrated by Fig. 813.   In this.

FIG. 242.
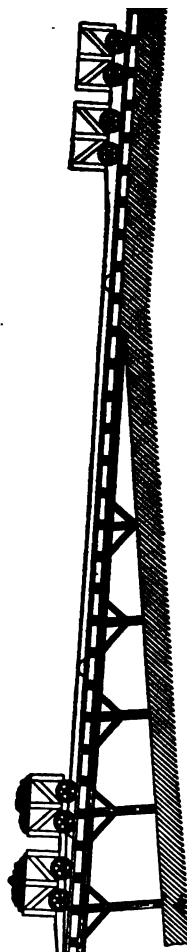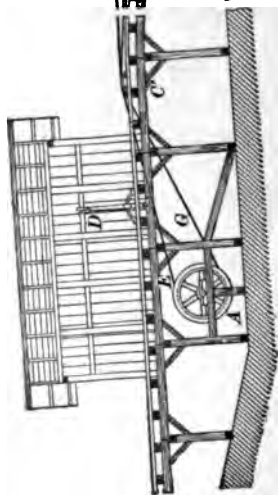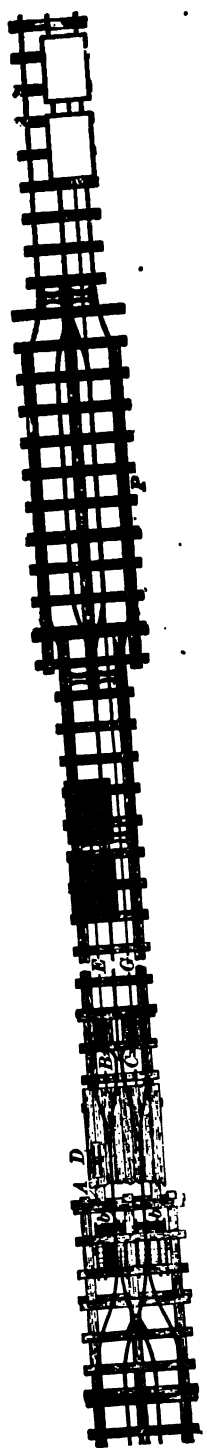
FIG. 818.
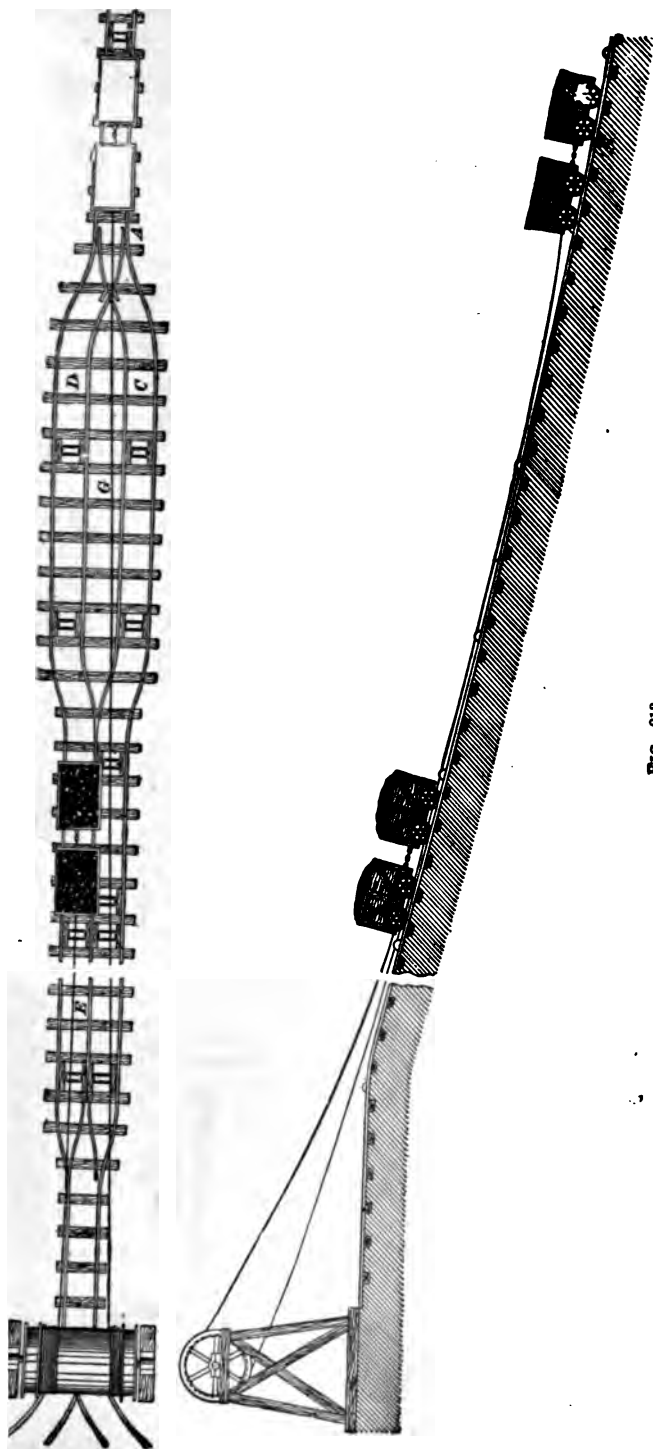
Fig. 817.
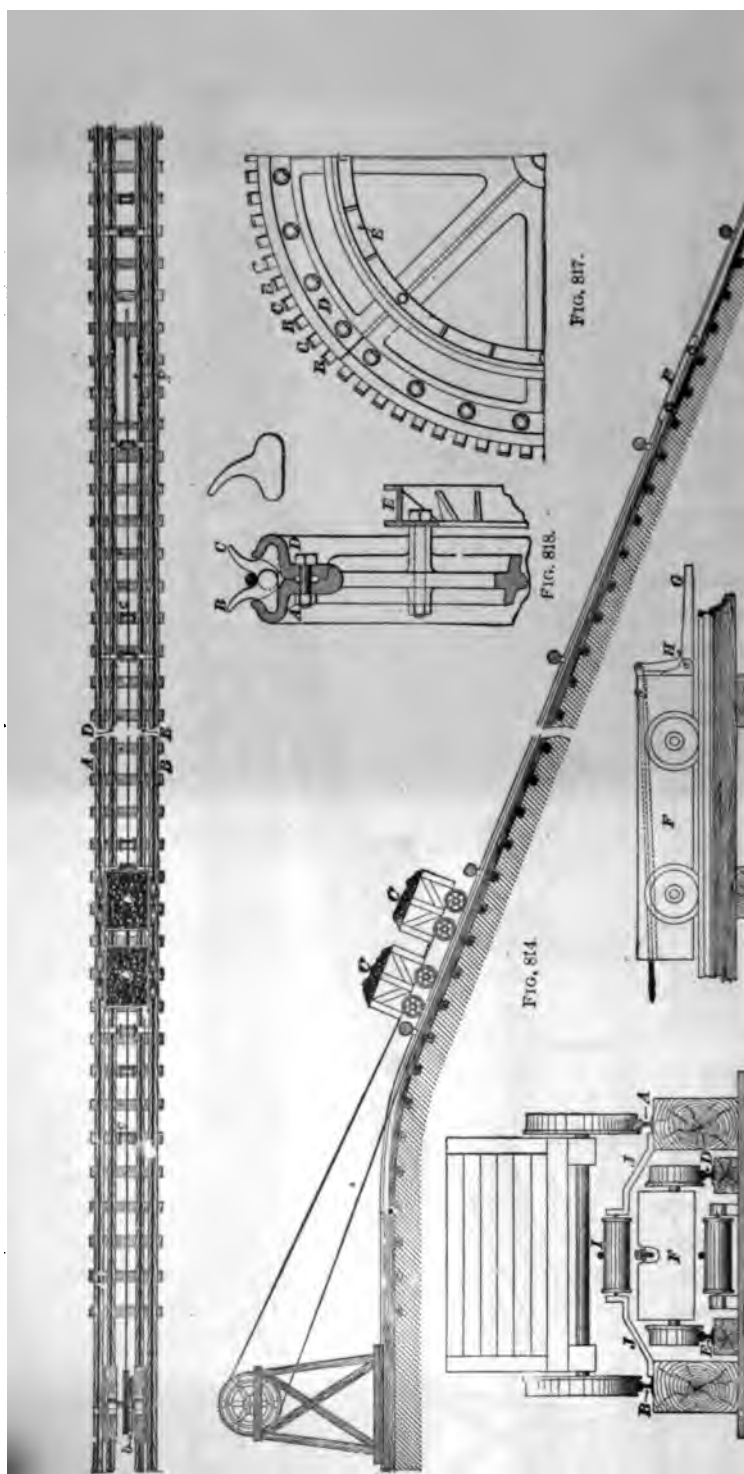
Fig. 818.

Fig. 814.

as in Fig. 812, the treble rails are continued to the part-ings, but the parting tracks unite in a single one, which is continued to the foot of the incline. These three fig-ures bring before us in a strong light some of the practi-cal difficulties that arise in making tracks for self-acting inclines. In mines where the roof and floor are tender, it is a costly and a difficult matter to keep the roads secure and in good working order, and so much is this the case, that the advantages of gravity-planes must often be disregarded when the conditions will not permit the construction of double tracks.

2310. Timbering should be avoided as much as possible on a self-acting incline, because, in the event of a rope break-ing and the cars running away and being derailed, the tim-ber is knocked out and the roof is let down, with the result that a stoppage of work occurs, and great expense is incurred in repairing the road.

2311. Figs. 814, 815, 816, 817, and 818 show some special appliances for self-acting inclines. For example, they show the self-acting jig, in which the full cars C, C are hauling up the balance-weight F. This consists of a cast-iron box run-ning on wheels on a track of small gauge within the wider gauge of the car-track. This may be clearly comprehended by reference to Fig. 815, where the car is seen passing over the balance-truck F. In Fig. 814 the loaded cars are shown descending and the balance-truck ascending. To fully com-prehend the principles of action involved in this self-acting jig, suppose that the cars have reached the foot of the incline. The man at the top applies the brake; that is, he holds the rope securely with the brake until the loaded cars are de-tached and a fresh train of empty ones is attached. Then he eases off the brake, and the balance-car is allowed to descend and pull the empty train of cars to the top of the incline. In this illustration the grip-wheel is mounted on a head-frame; but it often occurs in a mine that the grip-wheel is fixed under the level of the road at the head of the incline.

In Fig. 816 a side view of the balance-truck is seen at F, with the drop-bar G attached to the rope; or, more correctly, the rope is attached to the bell-crank lever at the back end of the balance-car in such a way that when the weight of the truck is hanging on the rope, the lever G is elevated, and in the event of the rope breaking or the balance-truck becoming detached, the lever G by its weight falls, digs into the ground, and prevents the truck from running down the plane.

2312. A front and end view of a grip-wheel is given in Figs. 817 and 818. In many cases a fleet-wheel, or a wheel with a slight conical tread, on which the coils of rope slightly slip down towards the flange on the lower side, is used instead of a grip-wheel. The grip-wheel, however, has a special advantage which makes it an excellent substitute for a double-rope reel. In case one side of the rope breaks, the grips on the tread of the wheel hold the other side of the rope secure, for the rope is seized by the grips in which it lies. As all the grips surrounding the wheel (as those shown at B, C, B, C) act independently of each other, the rope is held as securely as it would be by a reel on which was coiled separate ropes. The grips, it will be seen, are fixed on the periphery of the wheel like so many teeth; one of them is shown in transverse section at B, C, Fig. 818.

B and C are called the jaws of the grip. The rope presses in between them, and in doing so the jaws distend at the bottom and close at the top until the rope is held as in a vise. There are many varieties of jaw actions for grip-wheels, but this is a truly representative one. For self-acting inclines in mines where the pitch of the road is comparatively small and the length of the incline is short, head-wheels are sometimes laid in a horizontal position, under the road, at the head of the incline. In such a case, the diameter of the wheel is made about equal to the distance from center to center between the full and empty tracks. Such a wheel, however, is seldom a grip-wheel but the groove in which the rope runs is made in the form of an acute angle, so that

e rope fixes itself and grips on the sides of the two enclo-
ng flanges.   The wheel carries on one of its sides a brake-

inge, so as to keep the
inning cars under control.
ometimes, however, where
ie pitch is considerable,
eparate reels, connected by
ear-wheels, as shown in
ig. 819, are used to pro-
ide such a lead for the
ipes as will not tend to lift
ie cars off the tracks in
ming onto and leaving
e top of the incline. When
ey are set overhead, the
nning-on and the run-
ng-off ropes both come
om the under sides of
e drums or reels.   The
e-sided lead of the ropes
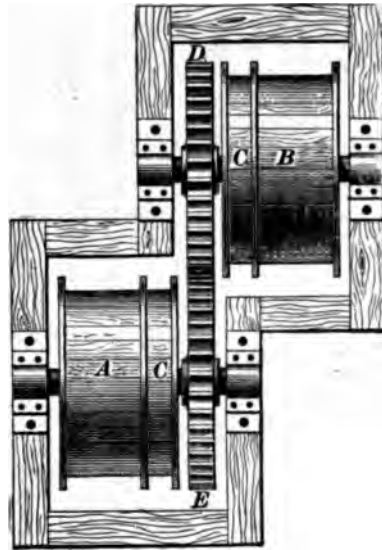
Fig. 819.

caused by the gear-wheels, for these make them turn
opposite directions.   These drums may be set under
e tracks at the head of the incline with great advan-
ge, if the pitch is considerable and the trains and ropes
e heavy.   Such an arrangement prevents the excessive
nding of the ropes in passing over the head-sheaves; that
the ropes running off the tops of the reels when set under
e tracks make a smaller angle with the line of the haulage-
e than when one of the ropes comes from the under side
the drum.   The advantage of these reels, then, may be
mmarized as follows: When they are set above the tracks,
e lead of the ropes is never too high, and when they are
under the tracks the lead of the ropes running on and off
e reels is never too low.

2313.   The necessity of using deflecting sheaves is
inly shown in Fig. 820.   Here two deflecting sheaves
ist be used, as shown at *F* and *E*.   The advantages of

FIG. 980.
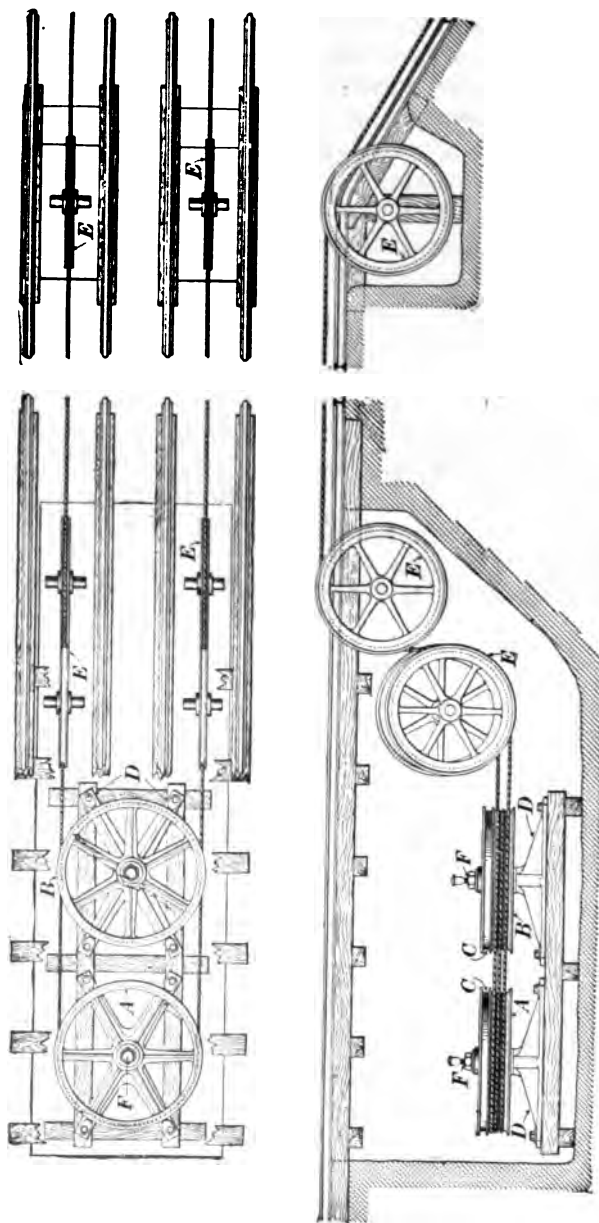
ioved sheaves, such as those at F, F, can not be doubted,
by them the trains on steep inclines can be held as se-
ely with the brake as they can be with drums; however,
damage to the ropes, in suddenly bending them round com-
atively small deflecting sheaves, must not be overlooked

314. The grooved wheels seen at the lower portion of
figure constitute a most ingenious device for holding
vy trains on an incline. To understand,
rever, the importance of the two grooved
els for holding or hauling, an explanation
heir construction and mode of action must
made. By referring to Fig. 821, it will be
1 that the tread of the periphery of the
el is made semi-elliptical in section, and the
= is made to run on at the high side of
curve r. After a complete coil has been
le, the running-on coil and the coil on the
of it begin to surge down onto a lesser
neter; consequently, the coils are always
ging over towards s, or, as the rope runs on,
eeps gently surging or fleeting downwards
il it at last begins to jam on the other side of the curve,
t o, the place at which the rope runs off. The disad-
tages of the fleet-wheel are, that the surging is not con-
ious, but intermittent, and the rope jumps and thuds and
cks, thus causing considerable wear and tear. When
rope runs on, it is arrested by the flange A, and presses
produces the surging or fleeting of the coil on the left
t. Sometimes, however, the intermediate coil slips down
the middle of the tread, and then a longer interval
ses before the coil r presses against the flange of the
el again, when another knock and surge takes place, and
n, continuously. From this, it is clear that it is impos-
= to run a rope into the grooves on a single wheel, for,
r two or three revolutions, the rope will roll up against
flange and jam there, so that the adjacent coils can not
: or surge away, and the action of the wheel will be
sted. Two wheels, however, can be used so as to take
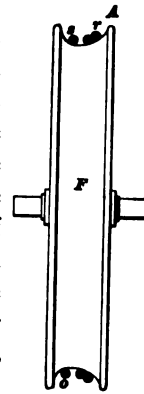
FIG. 821.

the place of one, by taking semicircular coils round each of them, or using the two wheels like belt-wheels.   This converts the coils that would otherwise envelop a single drum into belts on two drums in such a way that, if we denote the wheels by A, B, C, and A_1, B_1, C_1, respectively, the rope passes half round A, and from A to A_1; then from A_1 to B, and after passing half round B, to B_1, and after passing round B_1, to C, and after moving half round C, to C_1; finally, after passing half round C_1, the rope proceeds onwards to the haulage.

2315.   All that has been said so far relative to a description of the grooved wheels and fleet-wheels of an endless-rope haulage is important and worthy of attention, but there are other things that should be known, and not the least of these is the following:

When ropes are coiled around surfaces of any kind, particularly around wheels, what is called the *hold* or *grip* of the rope increases directly as the square of the number of coils.   If there are two coils on a fleet-wheel, the grip or hold of these two coils is 4 times that of one coil, and if there are three coils, the grip is 9 times that of a single coil. The same law holds true with the grooved wheels, for, although there are two wheels, three half coils on each wheel are equal to three whole coils on one.   Therefore, when the double wheels are connected by rope belts, as in this case, the friction obeys the same law as that of complete coils, and, consequently, the two wheels shown in Fig. 820, having three grooves each, secure a grip or haulage power of 9 times that of a single coil round one wheel.

2316.   The mechanical appliances for the successful working of self-acting inclines, such as reels, grip-wheels, rollers, switches, brakes, etc., etc., are important from a purely engineering point of view, and the best of these in the market can be purchased of manufacturers who make a specialty of them.   What is directly important to the mining engineer, as distinguished from the mechanical engineer, is the grading of the roads for self-acting inclines.   It is true that what is required to be done is of a comparatively

.ementary character, yet the principles involved must be
nderstood, to have their true value appreciated.

2317.   In Fig. 822 (*T*) the broken line shows a uniform
rade whose angle is equal to *B A C*.   It is possible that this
niform grade may be a complete failure, because the weight
f the descending coal may not be sufficient to lift the weight
if the long, heavy rope and overcome the friction due to the
lrum, rollers, rope, and cars, and yet the same incline might
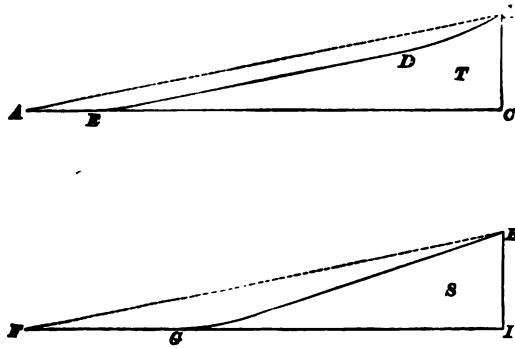
FIG. 822.

be made a complete success by so altering the grade as to
five it an increased fall at the top and a reduced fall at the
ottom.   For example, by altering the uniform grade *A B*
o the varying grade *A E D B*, the gravity power of
he coal is increased until the loaded train has reached
he point *D* and the empty train has reached the point *E*, at
'hich position they will have sufficient velocity or mo-
lentum to carry them to the point of parting.   Then, as the
'ains move on, the rope attached to the full cars will so
ngthen and the rope attached to the empty cars will so
orten as to make the rope not only move by its own
unterpoise, but to make it also assist the gravity power of
e coal.   Although the trains have acquired such a high
locity as to run the empty cars from *D* up to *B* and the
l cars from *E* to *A* by their inertia, the velocity of
: empty train, on reaching *B*, and the velocity of the full
e, on reaching *A*, is so low as to require no brake-power
unduly strain the ropes.   From this it will be noticed that

to make the grade of an incline successful where it other-
wise would fail, the inclination must be increased at the top
and reduced at the foot of the incline.

2318. In Fig. 822 (S) is shown a case where the fall
can not be increased immediately from the top, but the pitch
is sufficient to run the trains a considerable distance from
the foot of the incline G on a dead level. Very excellent
work may be done in this way. By lengthening the run,
the length and weight of the rope are correspondingly in-
creased. As the train leaves the top, it is prevented from
being unduly accelerated at the first portion of the run; as
the empty train must run a considerable distance from F to
G on a dead level, the momentum acquired becomes suffi-
cient to run the full train at the end of the run along the level
from G to F, and to run the empty train up to and over the
top H.

2319. The manifest drawback to the extension and
rapid action of self-acting inclines on small pitches is the
weight of the rope. If this factor could be eliminated, it is
easy to see that the incline could be prolonged indefinitely;
and it so happens that this can be done by applying the
principle of the endless-rope haulage. Inclined planes run
by an endless rope have been successful for very long dis-
tances on the surface, and in some cases in mines, but for
local or secondary haulage in certain places, jigging with a
balance-car will be found very effective.

2320. A description of the mechanical appliances
necessary in fitting up a self-acting incline is only important
to mining students in so far as their mode of action is con-
cerned. The principles of their construction belong to the
mechanical engineer; therefore, it is only necessary that
somewhat brief attention should be drawn to them. Be-
ginning at the head of the incline, attention is first called to
brakes. These are somewhat varied in construction, but
the mode of action of all is alike. There are only two general
varieties; namely, simple brakes for inclines of small pitch, on
which the cars never attain a very high velocity, and brakes
for holding greater loads on greater pitches, where the cars

:ain high velocities.   A brake of the former kind is shown
Fig. 823.   Its mode of action is that of a friction-block
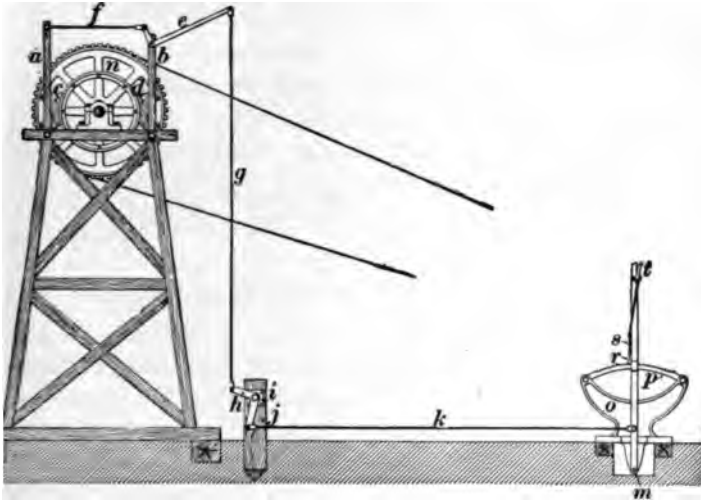
FIG. 823.

ı as is found on railway freight-cars.   The blocks *c, d* are
ched to upright levers *a* and *b*, and they are put into action

FIG. 824.

. series of levers and
:*e, g, h, j*, and *k*. The
r by which the brake
»plied is seen at *t m*.
; example is fairly
esentative of brakes
grip-wheels mounted
 standard frame. In
es, however, the grip-
el is usually fixed
erneath the tracks.
kes of greater hold-
 power are required
nclines of high pitch.
 a representative one
;hown in Fig. 824.
: flange of the brake-

wheel is seen at *A*, and is surrounded with the brake-strap which connects with the two short arms of the lever *B*. A brake of this kind is very powerful, and will hold securely a very heavy train.
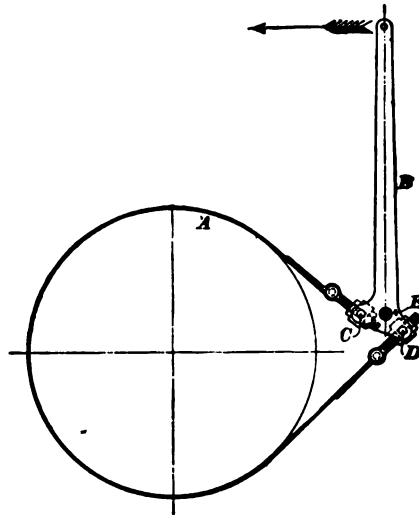
INCLINATION OF GRAVITY-PLANES.

2321. For gravity-planes to act with safety and economy, the following three important points must be considered:

1. Where the pitch is considerable and the length is short, only a minimum number of cars can be made to run in a train, or otherwise a heavy and expensive rope must be used, and powerful friction-brakes are required to hold the trains securely.

2. When the inclination is considerable, and the plane is a long one, the trains must be made larger to lift the heavy rope, whose increased weight is due to the relatively great length of the plane. Under these circumstances, a powerful brake is indispensable, because, after the loaded cars have passed the parting, the weight of the ropes adds considerable to the gravity of the coal, and, as a result, a powerful brake and a correspondingly heavy rope are required.

3. When the inclination of a gravity-plane is comparatively small, long trains are imperatively necessary; and even then it is necessary to provide, as has been previously shown, an increased fall from the top of the incline and a reduced grade at the foot of the incline, to run off the work with sufficient velocity.

2322. Gravity-planes have been sources of so much trouble and disappointment to those engaged in making them and using them, that it is necessary that the underlying principles should be understood to avoid such mistakes as sometimes are made, in the absence of proper knowledge, of how the earth's attraction becomes the operative force on self-acting inclines. This may be made very clear by a plain statement of facts.

First, the motive force is generated by something falling, and that something falling is the coal in the loaded cars.

Second, the force must do work in overcoming two kinds of resistances; namely, the friction common to the cars, rollers, sheaves, ropes, and coal, and the lifting of the weight of the rope attached to the empty cars at the commencement of the run.   The lift in the latter case increases with the length of the plane, and, as has been shown, the rope may become so heavy as to neutralize or counterpoise the weight of the falling coal, and thereby render necessary long trains, or the abandonment of the gravity-plane and the substitution of power haulage.

2323. It is essential that some simple, yet important, calculations should be made to determine when a gravity-plane will be safe and successful in doing the local or general haulage in a mine.   The calculations referred to should enable the student to find two results: first, the minimum number of cars in a train that will run with sufficient speed to do the required work, and, second, the maximum length and minimum pitch of a gravity-plane that will act efficiently.

Before proceeding, however, with an explanation of the methods of making the required calculations, it is necessary that the student should understand how the mechanical movements of the power and the work are related to each other.

2324. To understand this matter, the following general definitions are necessary:

1. *The forces on inclines are inversely proportional to the 'istances through which they move for a given amount of 'all.* For example, suppose a body has to move down an icline through a distance of 100 feet to fall a vertical dis-ince of 8 feet; then the force required to support this body 1 the incline is only equal to eight one-hundredths of its eight.   If the body weighs 80 pounds, the force is $\frac{8}{100} \times$) = 6.4 pounds.

2. *The forces on inclines are inversely proportional to the nes along which they act when those lines are parallel to 1e direction of the movements of the balancing bodies.*   For

example, in *A*, Fig. 825, the body or **weight** *L* **is** moving up or down the incline *a b*; therefore, **the** force **that** moves it is acting aiong the line *g h*, which is parallel to *a b*. Again, the balance-weight *W* is moving in a line *r s*, and this is parallel to the vertical line *b c*. Since *b c* is one-half of the length *a b*, if *L* weighs 120 pounds, *W* must weigh 60 pounds, because $\dfrac{b\,c}{b\,a} = \tfrac{1}{2}$. This is true when the velocities of *b c* and *b a* are equal, and when the force moving *L* is parallel to the direction in which *L* moves, assuming that the balance-weight *W* falls through a distance equal to that through which the load *L* moves along the incline.
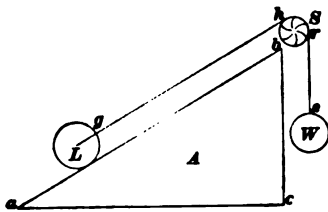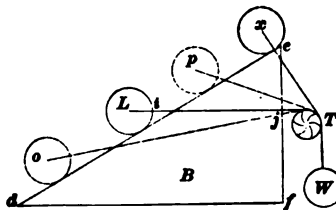
Perhaps matters of this kind are better understood by contrasts; therefore, suppose that the force moving the weight *L* does not act along a line that is parallel to the direction in which the load moves. In *B*, Fig. 825, the load is moving from *d* to *e*, and the force is then acting along the line *i j*. The force then acting along the line *i j* must be greater than the force acting along the line *g h* in the diagram *A*, because *d f* is shorter than *d e*, and we know that the forces in this case are inversely proportional to the lines along which they act. (See the *Inclined Plane* in Mechanics.) Again, if the load *L* is in the position *x*, the weight *W*, if infinite, can not move the load higher up the incline, because the force is acting through a cord whose direction makes a right angle with the direction of the incline. Therefore, an infinite force can not move the body, because the reaction of the incline is as great as the action of the force. It is thus seen that the forces on inclines must be taken as acting parallel to the lines along which the weights or forces act.

FIG. 825.

3. If the weights are equal, that is, if the weight moving along a vertical line is equal to the one moving along an ncline, to balance each other, their velocities must be pro-ortional to the lines along which they move; that is, the reight L, Fig. 825, A, will have to move from a to b in he same time that W moves from b to c.

A careful study of these three definitions will remove all erplexity concerning the balancing forces on an incline.

2325. Above all other considerations, there are two 1at stand out in bold relief in relation to gravity-planes. 'hese are:

1. The inclination must be sufficient for trains of reason-ble size to run off the work.

2. There must be a sufficient number of cars in a train on given incline to overcome the weight of the rope on a iven length of plane.

2326. It is first necessary to show how to find the umber of cars in a train under the given conditions. To iake the reasoning clear, let the following letters represent ie given and required values:

W_1 = the weight in pounds of the descending loaded car;
W_2 = the weight in pounds of the ascending empty car;
W_3 = the weight of the hauling-rope in pounds;
a = the percentage of the grade expressed decimally, which is the same as the sine of the angle of in-clination;
$\frac{1}{40}$ = the coefficient of friction;
W = the required number of cars in a train.

The force required to overcome the resistance due to the eight of the rope is equal to $a\,W_3$.

The force required to overcome the resistance due to the pe, rollers, and drums is equal to $\dfrac{W_3}{40}$.

. Denoting the total force required to overcome the weight and friction of the rope by F_2, we have

$$F_2 = a\,W_2 + \frac{W_2}{40}. \qquad (197.)$$

EXAMPLE.—A rope 2,000 feet long weighs 4,000 pounds, and the inclination of the plane on which it is used is equal to a grade of 8 per cent. What force is required to move the rope ?

SOLUTION.—Applying formula **197**, we have

$$F_2 = a\,W_2 + \frac{W_2}{40} = (.08 \times 4,000) + \frac{4,000}{40} =$$

420 pounds, the tractive force required.

2327. To find the available gravity force for overcom— ing the required tractive force of the rope, observe that on a self-acting incline, for every load of coal two cars are re— quired, and as the cars and load can not move without being subject to the resistance of friction, the following equation expresses the amount of this resistance:

$$\frac{W_1 + W_2}{40}.$$

Again, as the cars balance each other on an incline, nothing falls but the load, but all of the load is not available for overcoming the tractive force required to move the rope, for the friction due to the cars and the load must be sub— tracted from the gravity force of the coal, in order to find how much force each pair of cars can supply to move the rope.

Denoting the total gravity force due to the coal by F_1, we have

$$F_1 = a\,(W_1 - W_2). \qquad (198.)$$

Again, denoting the available gravity force due to the coal by F, we have

$$F = a\,(W_1 - W_2) - \left(\frac{W_1 + W_2}{40}\right). \qquad (199.)$$

EXAMPLE.—A gravity-plane has a grade of 8 per cent.; it is 2,000 feet in length, the rope attached to the empty cars at the foot of the

ncline weighs **4,000** pounds, a loaded car weighs **4,000** pounds, and an mpty one weighs **1,800** pounds. What is the number of cars that ust run in a train to overcome the resistance of the rope at the start f the run ?

SOLUTION.—Applying formula **197,** we have

$$F_2 = a\, W_2 + \frac{W_2}{40} = (.08 \times 4{,}000) + \frac{4{,}000}{40} =$$

) lb., the force required to move the rope.

Applying formula **199,** we have

$$= a(W_1 - W_2) - \frac{(W_1 + W_2)}{40} = .08\,(4{,}000 - 1{,}800) - \frac{(4{,}000 + 1{,}800)}{40} =$$

31 lb., the available gravity force due to one pair of cars.

Therefore, the number of cars that must run in a train is equal to

$$\left(a\, W_2 + \frac{W_2}{40} \right) \div \left[a\,(W_1 - W_2) - \frac{(W_1 + W_2)}{40} \right] = \frac{420}{31} = 13.54 + ;$$

or, 14 cars in a train.   **Ans.**

EXAMPLE.—The grade of an incline is 7 per cent., the length of the ine is 2,000 feet, the weight of the rope is 4,000 pounds, the weight full car is 4,000 pounds, and that of an empty one is 1,800 pounds. w many cars must there be in a train for the plane to be self-acting ?

SOLUTION.—Applying formula **197,** we have

$$F_2 = a\, W_2 + \frac{W_2}{40} = (.07 \times 4{,}000) + \frac{4{,}000}{40} =$$

lb., the tractive force required for the rope.

Applying formula **199,** we have

$$= a(W_1 - W_2) - \frac{(W_1 + W_2)}{40} = .07\,(4{,}000 - 1{,}800) - \frac{(4{,}000 + 1{,}800)}{40} =$$

9 lb., the available force of the load.

Therefore, the number of cars in a train will be equal to $\frac{180}{9} = 42\frac{2}{9}$, 3 cars.   **Ans.**

EXAMPLE.—The grade of an incline is 6.6 per cent., the length of road is 2,000 feet, the weight of the rope is 4,000 pounds, the ;ht of a loaded car is 4,000 pounds, and that of an empty one is) pounds. How many cars must there be in a train for this incline e self-acting ?

SOLUTION.—Applying formula **197,** we have

$$F_2 = a\, W_2 + \frac{W_2}{40} = (.066 \times 4{,}000) + \frac{4{,}000}{40} =$$

b.. the tractive force required for the rope.

Applying formula **199**, we have

$$F = a(W_1 - W_2) - \frac{(W_1 + W_2)}{40} = .066 \times (4,000 - 1,800) - \frac{(4,000 + 1,800)}{40} =$$

.2 lb., the available gravity force due to the load.

Therefore, the number of cars in a train is equal to $\dfrac{364}{.2} = 1,820$ cars. Ans.

It is plain, however, that such a number could not be made to act in practice, for observe the absurdities involved in a case like this. The plane is only 2,000 feet in length, and if the cars were each 7 feet long, then the length of a train of cars would be equal to $1,820 \times 7 = 12,740$ feet, or a train would be $\dfrac{12,740}{2,000} = 6.37$ times the length of the incline.

2328. The self-acting incline that must next engage attention is the jig system. As has been shown, the jig or balance-carriage runs on a narrow-gauge track within the car-track, or it is made to run in a parallel opening. The weight of the jig is equal to that of an empty car plus half the weight of the coal a full car carries. The result is, that when a loaded car descends the incline, only half the weight of the coal it carries is available for gravity force, because the other half of the weight of the coal does the gravity work of raising the jig. The excess of weight in the jig does the work of raising the empty car. As only one-half of the weight of the coal does gravity work during the descent of the coal, and the other half does work in the hoisting of the jig, it is only on inclines of high pitch and relatively short length that the jig system can be adopted.

EXAMPLE.—The grade for a jig incline is 20 per cent., and the length of the road is 200 feet. The weight of the rope per foot of length is 1.2 pounds, the weight of a full car is 4,000 pounds, the weight of an empty one is 1,800 pounds, and the weight of the jig is 2,900 pounds. Prove that this jig incline can not be self-acting.

SOLUTION.—Applying formula **197,**

$$F_2 = a W_3 + \frac{W_3}{40} = .20 (200 \times 1.2) + \frac{200 \times 1.2}{40} = 54 \text{ lb.,}$$

the resistance to be overcome in moving the full length of the rope.

Consider the conditions existing when the loaded car is at the top and the jig is at the bottom of the plane. The jig can be considered as an empty car weighing 2,900 lb.; applying formula **199**,

$$F = a(W_1 - W_2) - \frac{(W_1 + W_2)}{40} = .20(4,000 - 2,900) - \frac{(4,000 + 2,900)}{40} =$$

47.5 lb., the available gravity force at the descent of the full car.

Therefore, the jig incline can not be self-acting, because there are only 47.5 pounds of gravity force available to overcome a resistance of _ pounds.

One thing, however, can be done, and that is, a level run of 30 feet can be made at the foot of the incline, which will give the loaded car on the one hand and the jig on the other sufficient force to overcome the initial and succeeding resistance. For, at the start of the run, the jig or the empty car does not offer any gravity resistance, since it is on the level run. In consequence, the gravity force is increased in the following large proportion: The weight of the jig is 2,900 pounds; therefore, a force of .2 × 2,900 = 580 pounds is required, apart from the traction due to friction, to move it up an incline having a 20 per cent. grade. Here, then, can be seen the great advantage due to the right method of making a level at the foot of an incline to neutralize the resistance due to the weight of the rope at the beginning of the plane. As stated before, the weight of a jig is usually equal to the weight of the empty car plus one-half the weight of the coal it can carry.

EXAMPLE.—A self-acting jig incline is in all respects the same as the last, excepting the grade, which is in this case one of 10 per cent. Prove that it will be self-acting when a short level is provided for the start of the jig or empty car before it begins its ascent.

SOLUTION.—Applying formula **197**,

$$F_2 = aW_2 + \frac{W_2}{40} = .1 \times 240 + \frac{240}{40} = 30 \text{ lb.} =$$

the resistance to be overcome in moving the rope.

Applying formula **199**,

$$F = a(W_1 - W_2) - \frac{(W_1 + W_2)}{40} = .1(4,000 - 2,900) - \frac{(4,000 + 2,900)}{40} =$$

10 − 172.5 = − 62.5 lb. = the available gravity force when the grade is uniform to the bottom of the incline. This negative result shows that the incline can not act. But if the jig commences its journey for the

· ascent of the incline along a level, the resistance due to the ascent of the jig is reduced by .1 × 2,900 = 290 lb.; hence, (110 + 290) − 172.5 = 227.5 lb., and as the resistance of the rope due to gravity and friction is only 30 lb., the self-acting jig incline will act most efficiently, because, after allowance has been made for all resistances, there remains an excess of force equal to 227.5 − 30 = 197.5 lb.

2329. The determination of the size of a rope for the working of a self-acting incline is best done by first assu· ming that a rope of a given size and weight will answer, and then finding the tension due to the movement of the rope. If the tension found is too much for the rope, then a heavier one must be assumed and tested.

EXAMPLE.—Required the tension in a rope under the following conditions: A loaded car weighs 4,000 pounds, an empty one weighs 1,800 pounds, there are 10 cars in a train, the grade is one of 8 per cent., the length of the incline is 1,900 feet, and the weight of the rope per foot of length is 1.2 pounds. What is the tension in the rope at the moment the loaded trip leaves the top of the incline?

SOLUTION.—Let T equal the tension in the rope. Then,

$$T = \frac{(10\,W_2 + W_3)}{40} + a(10\,W_2 + W_3).$$

The weight of the rope is equal to $W_3 = 1,900 \times 1.2 = 2,280$ pounds.

Then, $T = \dfrac{(10 \times 1,800 + 2,280)}{40} + .08\,(10 \times 1,800 + 2,280) = 2,129.4\,\text{lb.}$ Ans.

As the working load of this plow-steel rope is 7 tons, it appears to be too heavy, but really the jerking strains to which such ropes are subjected render it necessary that the rope should be 7 times as strong as the calculated quiet load.

AUTOMATIC SWITCHES.

2330. Every gravity-plane on which the descending loaded cars raise the empty ones should be provided with an automatic switch at the head of the plane. There are two general types of switches for this purpose; both are called automatic switches, although, strictly speaking, only one is automatic, the other requiring the attention of the runner. In Fig. 826 is shown an automatic switch consist· ing of three tongues a, b, and c, pivoted ·to the rails as shown. To the tongue a is hung a weight w, in

sequence of which it is always kept closed; i. e., kept
inst rail *d*. With the tongues arranged as shown
he figure, the loaded cars coming from track *A* force
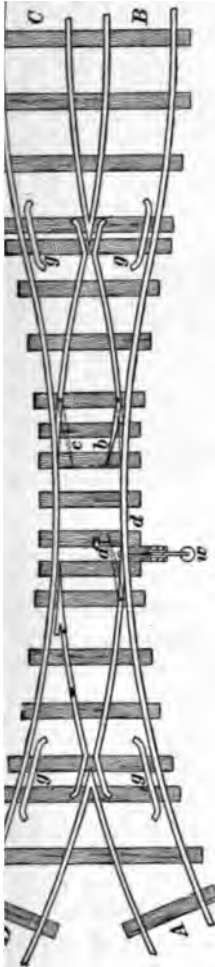
open the tongue *a*, as shown by the
dotted lines, and pass over the track
B, where they may be let down the
incline, the tongue being closed by
the weight *w* after the cars pass over.
The cars coming up the slope on
empty track *C* open the tongue *c* and
close tongue *b*, as shown by the dotted
lines. These tongues always remain
in the position placed by the last train
of empty cars run over them. On the
next trip, the loaded cars from track
A pass over to track *C*, where they
may again be let down the slope, and
the empty cars coming up the slope
on track *B* move the tongues *b* and *c*
to their original position, and pass
on to track *D*. The above opera-
tion is then repeated, the loaded cars
taking alternately the tracks *B* and
C, and the empty ones always the
track *D*.

2331. In those gravity-planes in
which three rails are used from the
head of the slope to the parting,
and only two from the end of the
parting, an automatic switch must
be placed at the junction where
the two rails unite with the parting.

FIG. 826.

In Fig. 827 is shown an automatic
ch, which may be used at such a place. Here two
ers *A* and *B*, pointed at the ends and bound with iron,
pivoted at *C* and *D*, respectively, in such a manner
they may move freely over the tops of the rails.

With the timbers in the position shown, the empty cars coming along track E will be pulled up the slope on track F, while the descending loaded cars coming down the slope on track G will shift the timbers over the rails to the position shown in dotted lines, and go along track E. At the next trip the empty cars coming along track E will be pulled up the slope on track G, while the descending loaded cars coming down the slope on track F again shift the timbers to their original position. The above operation is then repeated, the empty cars being pulled alternately up the tracks F and G. Blocks of wood or iron a, a, a are securely fastened to the ties, to prevent the timbers from moving too far; the timbers being thus blocked, serve for guide-rails, to guide the wheels of the cars to their respective tracks.
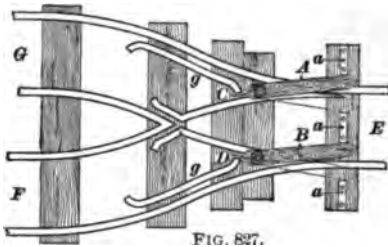
FIG. 827.

SAFETY-BLOCKS.

2332. At the head of the slope near the brink on all gravity-planes should be placed an arrangement called a **safety-block**, for preventing the cars from descending the plane before they are properly attached to the rope. There are various forms of these blocks in use, differing in construction of their details, but representing only one principle in all; namely, that of providing an obstruction, either over the rails or in the center of the track, to prevent the cars from passing, and of removing this obstruction when the cars are to be let down the slope.

2333. One of the best forms of safety-blocks is shown in Fig. 828, in which A and B are two timbers pivoted at C and D, respectively. The end of each timber is shaped as shown, and iron-bound. Directly in the center between these timbers is fastened an iron plate E having a slot in it through which the vertical part F of the rod G may be moved back and forth. The timbers A and B are connected

by two wrought-iron links *H* and *I*, which form a toggle-joint, as shown. The ends of these links, meeting in the center, are fastened to the end of the rod *F* projecting up through the slot in the plate *E*. *J*, *J* are wrought-iron levers placed on the outer sides of the track close to the rails. These levers are pivoted at *K*, and are connected together by a rod *L*, at the center of which the rod *G* is so fastened that when either of the levers is moved the other moves with it. The operation of this arrangement may be explained thus: With the timbers in the position shown, a train of loaded cars coming along track *M* takes track *N*,
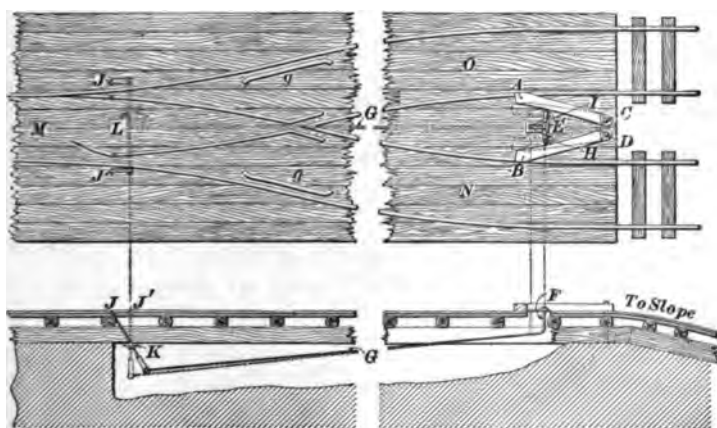
FIG. 828.

and goes along until the wheel on the inner rail of the front car strikes the timber *B*. After the rope has been attached to the cars, the runner shifts either one of the levers *J* to the position *J'*, shown by the dotted lines. In doing this, the rod *G* is pulled to the left, the vertical part *F* of it sliding in the slot of the plate *E* to the left also, which causes the ends of the links *H* and *I* fastened to it to go with it, thereby causing the blocks *A* and *B* to take the position shown by the dotted lines. The tracks being thus freed, the cars may be let down the slope. The empty cars coming up the plane on track *O* pass along until the first car reaches the

lever *J* on the outside of this track, which is now in the position *J'* shown by the dotted lines, and moves it to its original position *J*, in passing to track *M*, thereby again placing the timbers *A* and *B* over the tracks.   On the next trip, the loaded cars coming along track *M* take track *O*, and are prevented from descending the slope by the timbers being placed over the inner rails by the last train of empty cars. After the rope has been fastened, the timbers are again moved to the dotted position by one of the levers *J*, and the cars may then be let down the slope.   The empty cars coming up this time on track *N* pass along until the front car reaches the lever *J* on the outside of this track, which is again in the position *J'* shown by the dotted lines, and moves it to its original position in passing to track *M*, thereby again placing the timbers *A* and *B* over the tracks. This operation is repeated every trip, the empty cars coming up the plane and automatically placing the timbers over the rails, to prevent the loaded ones from descending the plane.

2334. In Fig. 829 is shown another good form of a safety-block, consisting of a heavy wrought-iron bar *A* firmly keyed on a shaft *B*, that is held in position by the bearings *C*, *C*, which are bolted to suitable supports.   The top of the front end *a* of the bar is inclined, as shown in the figure, and is caused by the weight *W* to project up in the center of the track to such a height that it will strike the axle of the cars, this height being governed by the timber *H*.   At one end of the shaft *B* is keyed a lever *D* (placed at one side of the track) by which the block is operated. One of the blocks is placed in the center of each track. With the tongues of the switch in the position shown, the loaded cars coming along track *E* take track *F*, and run along it until the front axle of the first car strikes the projecting part *a* of the bar *A*.   After the rope has been attached to the cars, the lever *D* is pulled to the right, which causes the projecting part *a* to swing down, when the cars may be let down the slope.   The projecting part *a* should be held down until all the cars have passed over it

r *D* is then released, and the part *a* is again **brought**
ts proper height by the weight *W*. The train **of**
ars coming up the slope on track *G* finds the block
position shown, the axles forcing the projection *a*
·hich may be readily done, since it is inclined, and
in passes over to track *E*. After the cars have
»ver it, the projection *a* is again brought up by the
W to its original position, as shown. On the next
loaded cars coming along track *E* will run along
until the front axle of the first car strikes the pro-
z, and may then be lowered, after the ropes have
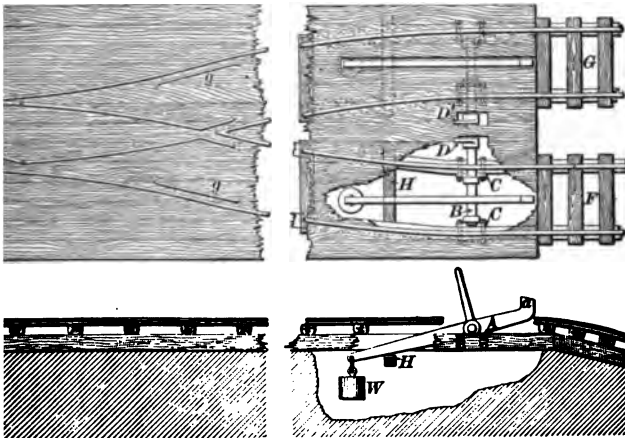
FIG. 829.

tened to them, by pulling the lever *D'* to the right,
ling it until the last car has passed down the plane.
pty cars this time, after coming up the plane on
and depressing the projection *a* as before, pass along
, after which the projection *a* is again brought to
nal position by the weight *W*. This operation is
eated, the loaded cars being let down the slope alter-
n each track by pulling either the lever *D* or *D'*, and
ty cars coming up the slope depressing the projec-
When this block is used, it is impossible for the cars
at the head of the slope to run down the plane
the will of the operator, since they are always in the
osition to prevent the cars from passing.

2335. In Fig. 830 is shown another safety-block which may be used on gravity-planes where light loads are run. This consists of two iron-bound timbers *A* and *B*, pivoted at *C* and *D*, respectively, in such a manner that the timber *A* can be swung over the top of the rail. One of these blocks is used for each track. With the timbers in the position shown, a loaded car coming along track *E* will be prevented from descending the plane. After the rope has been fastened to the cars, the timber *B* is swung to one side, so as to allow *A* to take the position shown by the dotted lines. The empty cars coming up the plane on track *F* find the
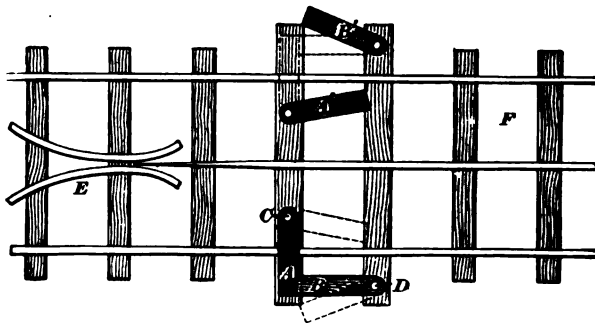
FIG. 830.

timbers in the position shown on that track, and pass along, after which the timber *A'* is swung over the track, and is locked by the timber *B'*, as shown by the dotted lines. On the next trip the loaded cars coming along track *F* this time find the track closed. After the rope has been fastened to the cars, the timber *B'* is swung over, and the cars are let down the slope, the timber *A'* being moved by the wheels to its original position. The empty cars coming up the slope on track *E* find the timbers *A* and *B* in the position shown by the dotted lines, and pass along, after which the timbers *A* and *B* are again placed in their original position by the runner. This operation is then repeated, locking and unlocking the blocks on each track alternately.

2336. At the foot of the gravity-planes before described there should be a slightly inclined surface for the

:eption of the cars after they have descended the plane,
ie cars being prevented from running along the surface by
rragging the wheels.   If they are to be run in a tipple,
e first car may be uncoupled and the sprags removed,
:reby letting the car run along the track by gravity to
: tipple.   Instead of doing this, the safety-block shown
Fig. 831 may be used, which is entirely similar to that
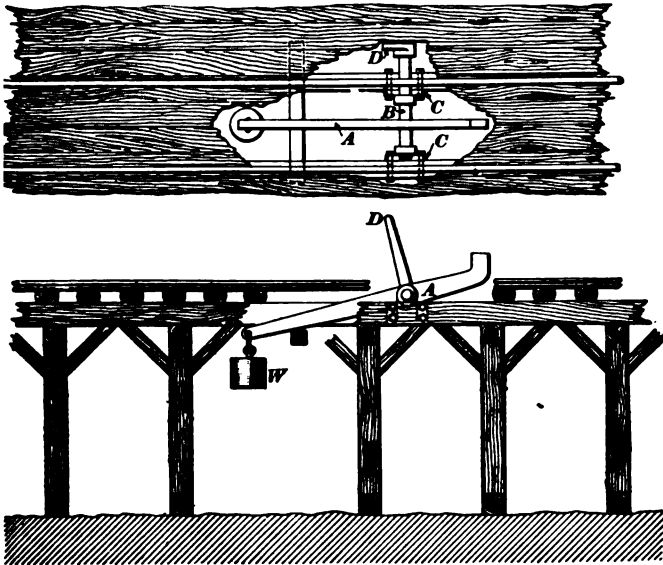
FIG. 831.

strated in Fig. 829, except that the short end of the lever
is not inclined as shown in Fig. 829.   With this arrange-
nt, after the cars have descended the plane and been
agged, they are uncoupled and the sprags are removed,
reby letting them run along the track by gravity until
front axle of the first car strikes the projecting part of
bar A.   When a car is to be run on the tipple, the lever
s pulled to the right, thereby swinging the bar A, the
jecting curved part clearing the axles, and the car passes.
er the rear axle has passed over the projecting part of the
A, the lever D is released, and the weight W falls down
raises the projecting curved part to its original position

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before the axle of the next car strikes it. With this arrange-
ment, the cars can be allowed to run to the tipple as required,
it being very easy to handle, and simple in construction.

2337. As a safeguard against life and property, every
inclined plane should be provided with some kind of an
arrangement to prevent the cars from descending to the
bottom of the slope in case the rope breaks. The arrange-
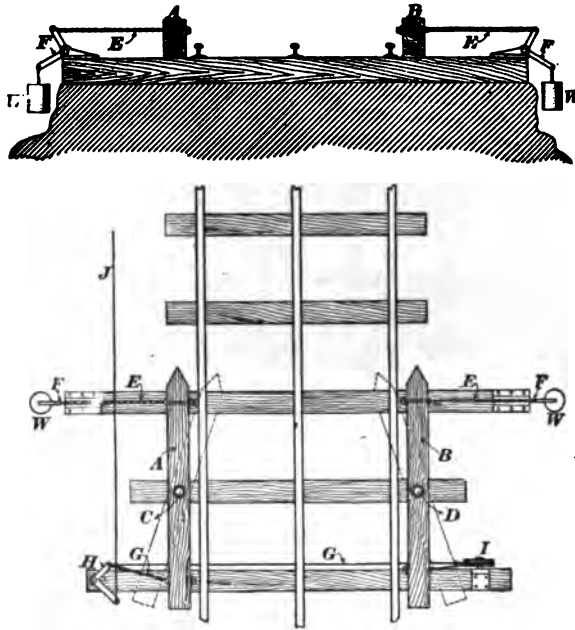
FIG. 832.

ments in general use for this purpose are very crude, either
stopping the cars coming down the slope, or switching them
off and throwing them from the track. In Fig. 832 is
shown an arrangement called a **safety-lock** used for stop-
ping the cars. This consists of two timbers *A* and *B* placed
on the outside of the track, having their front ends pointed
and iron-bound, and pivoted at *C* and *D*, respectively, in
such a manner that the pointed ends may be swung over
the rails. At the front end of each timber is fastened a
chain *E*. This chain is also connected to one leg of the

l-crank *F*, to the other leg of which a weight *W* is hung,
ich causes each timber to always take the position shown
:he figure.   To the other end of each timber are fastened
chains *G*.   Each end of these chains is connected to one
of the bell-crank *H*, the chain which is fastened to the
ıber *B* being led over the wheel *I*, which is securely
tened to the ties, and then led under the rails to the bell-
nk *H*.   To the other leg of the bell-crank *H* is connected
ıire *J*, which is led to the head of the slope, where it
y be pulled by the runner.   The operation of this may
explained thus: Upon the rope breaking and the cars
ning down the slope, the runner at the head of the plane
ls the wire *J*, which causes the bell-crank *H* to swing the
ıted ends of the timbers *A* and *B* over the rails into the
ition shown by the dotted lines.   The wheels of the cars,
n reaching this point, strike the timbers, and a general
ısh-up follows.   It can well be supposed that this lock
st be repaired after each time it has been in use.

338.   An arrangement for switching the cars off the
:k is shown in Fig. 833, in which two tongues *A* and *B* **are**
:ed as shown and fastened to a
ın *C*, one end of which is con-
:ed to one leg of the bell-crank *D*,
ıng a weight *W* hung to the
ır leg, which causes the tongues
ılways take the position shown.
. other end of the chain *C* is led
ınd the pulley *E*, which is fast-
1 to the cross-tie, and is at-
ıed to a wire *F* led to the head
:he plane.   This arrangement
:its of the empty cars being run
he slope, since the wheels force
· the tongues *A* and *B* to the
tion shown in dotted lines.
ın the loaded cars come down
slope, the tongues *A* and *B*
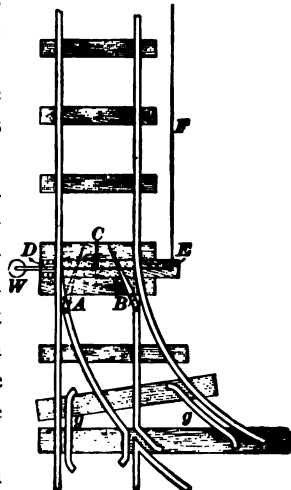
FIG. 833.

must be pulled over by the wire F to the position shown by the dotted lines, so that the cars may pass. In case the rope breaks at any point above the switch, the wire F is not pulled, since the tongues are always in position, or closed, so that the runaway cars will be switched off to one side. This arrangement possesses the property of always being set to switch the runaway cars off the track; but this is done at the expense of an extra amount of labor on the part of the runner, as he must pull the wire F, in order to open the switch when the descending cars reach it, for, otherwise, they would be switched to one side.

ENGINE-PLANES.

GENERAL DESCRIPTION.

2339. This is a system of haulage that is adopted on inclined roads where the pitch is just sufficient to run the trains down grade with the hauling-rope attached, or where
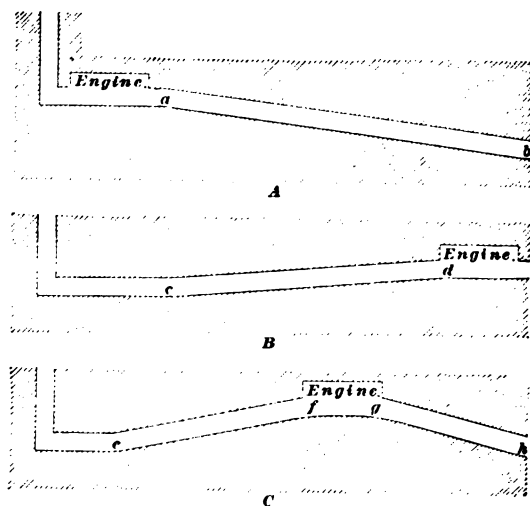
Fig. 834.

the direction of the pitch is such that the loaded cars must be hauled up grade. By reference to A, Fig. 834, the direc-

tion of the pitch is such that the trains must be hauled up grade, whereas in *B* the loaded cars must be lowered, because the direction of the pitch is down grade. Hence in *A* the hauling-engine is located at the shaft, whereas in *B* the engine is located either at head of the incline or at the shaft. In the latter case, the rope is conducted along one side of the track, and carried round a return sheave, as shown in Fig. 835. In *C*, Fig. 834, the hauling is done on two reverse inclines; consequently, the engine must be placed at the highest elevation of the inclined roads.

2340. Fig. 834 shows three distinct classes of engine-planes: (1) Those on which the loaded cars are hauled up grade by the engine, and the empty cars are run back by gravity, as in *A*, where the full trains are hauled up grade from *b* to *a*, and the empty cars are run back from *a* to *b*. (2) Those on which the loaded train runs down an easy grade, hauling the rope with it, and where the empty train must be hauled up grade with the engine. (3) Those on which the engine is located at the head of two reversely inclined roads, as in *C*. In the latter case, the engine hauls the loaded trains up grade from *h* to *g*, and then the loaded trains proceed down grade by gravity from *f* to *e*, and the empty trains are hauled up grade from *e* to *f* by the engine, and then run down grade from *g* to *h* by gravity.

2341. In mine haulage, engine-planes of the character shown in *C*, Fig. 834, are found to furnish the best possible results, for where the seam is undulating the reverse inclines are found to supply excellent conditions for long haulage to be done cheaply and expeditiously, because the engine can be located at the highest point between the two inclines. If the run from the shaft to the engine is a mile, and that from the engine to the foot of the incline *g h* is half a mile, one drum and one rope can be made to run the empty cars first from the shaft to the engine, and, second, to lower the cars from the engine to the foot of the down-grade incline *g h*. In short, a pair of reverse inclines can be made to obviate the necessity of the use of a tail-rope.
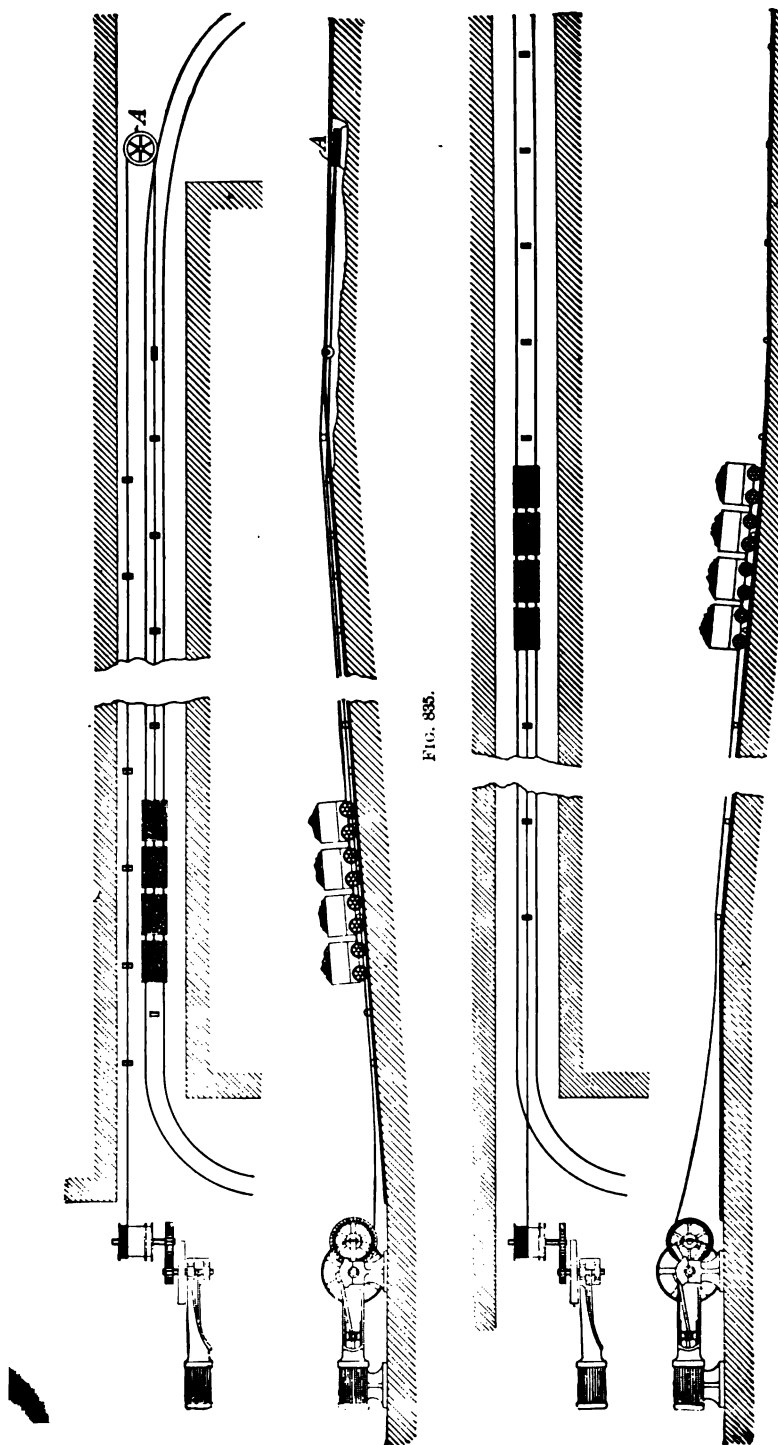
FIG. 835.

2342. There are two cases in which the engine-plane is superior to all other systems. They are: (1) where the seam is pitching heavily from the shaft, for then no type of locomotive can be used to do the haulage as cheaply and quickly; (2) when the road passes over two reverse inclines, where, however, the pitch from or to the shaft is small or just sufficient to run the train back with the rope; then locomotive haulage can sometimes be adopted with better results. Fig. 836 is a good illustration of an engine-plane haulage to the shaft, and shows in plan where the engine is located with reference to the lead or line of the rope.

2343. In some of the later installations of engine-plane haulage, the engine is not located in the mine, but on the surface, and the haulage-rope is conducted down the shaft, or down a bore-hole made for the purpose. Fig. 837 supplies a good illustration of how the hauling-engine may be located at the top of an incline for upward haulage; but this is a surface arrangement, and has attached to it an appliance that is seldom required in a mine. However, as it is used in connection with mine-surface appliances, and sometimes on slopes, it is here considered worthy of notice.

The device in question is the barney or truck M seen behind the full car that has just arrived at the top of the engine-plane. The barney is a little car that runs on rails set between the rails of the coal-car. The rope is attached to the barney, which is thus used to push the full car up the incline in front of it, so that when the full car reaches the top of the incline, it can run away by means of its inertia. Again, when the empty car reaches the foot of the incline at P, the barney dips down into the little pit at N, and becomes disengaged from the empty car, which, by the inertia acquired by the velocity due to its descent, runs into a parting to allow the next full car to run over the barney. When the engine starts, the barney rises out of the pit and bumps against the full car as before.
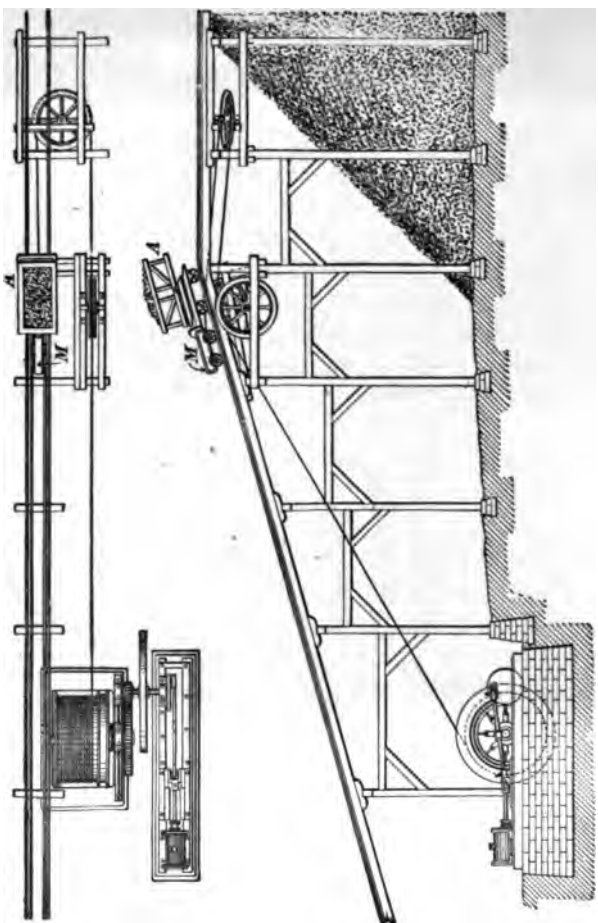
FIG. 887.
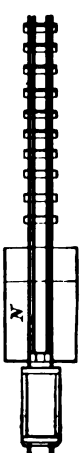
2344. Before considering the numerical calculations concerning engine-planes, some other matters of detail in the working of the tracks must be noticed. For example,
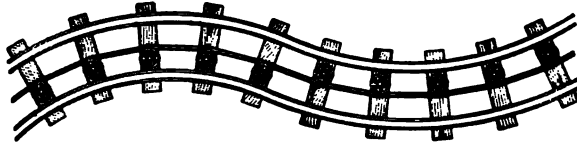
FIG. 838.

where curves occur the guide-sheaves for the rope are so set within the rails that the rope is made to run in the middle of the track, as in Fig. 838; or, if it is desirable to use larger
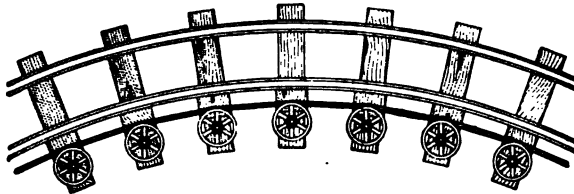
FIG. 839.

sheaves that will not damage or strain the rope by sharp bends, the sheaves are sometimes set on the side of the track that corresponds to the inside of the curve, as in Fig. 839.

2345. The drag-bar, or back-set, shown in Figs. 840 and 841, is a provision made for safety during the ascent of heavy trains on engine-planes.

In the event of a broken rope, this prevents the train
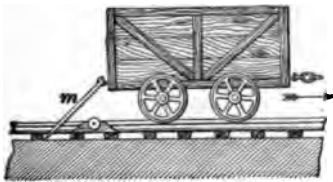
FIG. 840.
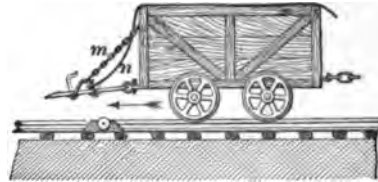
FIG. 841.

from running back, and by this means the damage that would otherwise occur is prevented.

A loose drag-bar is shown in Fig. 840. It is simply a strong iron bar hooked to the rear of the last car in the

train. Its mode of operation is as follows: As the train runs up the incline, the bar trails over the ground. Should the rope or one of the coupling-chains break, it sticks into the ground and prevents the train from running back. The drag-bar shown in Fig. 841 is the same in character as that shown in Fig. 840, except that it is prevented from trailing and knocking against the rollers on the ascent of the train, being suspended by the chain *M*, one end of which is attached loosely to a bent hook, which has the shape shown by the dotted lines. A second hook fits over the end of the first one, and the weight of the bar straightens the chain which supports it by means of the two hooks. In case of accident, the bar is dropped to the ground by the train-rider pulling the rope which is attached to the smaller hook on the right. This pulls the smaller hook over the projection on the end of the bent hook, thereby causing the latter to slip through the end link of the chain, and take the position shown by the dotted lines. This action releases the bar, which thereupon falls and digs into the ground.

2346. In a large mine there can not be a main haulage for which all the loaded cars are gathered at one station. Since this is the case, not only must the system of the main haulage be modified so as to run the work off from different stations, but sometimes a system that appears in all respects the best for the grades of the main haulage-roads must be abandoned for another that will allow the gathering-up stations to be located nearer to the working faces. Fig. 842 furnishes such a case. Here the head of the engine-plane is at the shaft, and the gathering-up stations are located at the entrance to the side entries. If the latter are driven along the strike of the seam, as they happen to be in this case, the main haulage is made to reach no farther than where the main rope is taking hold of the two full cars at *B*. It is clear, then, that the engine-plane haulage here adopted is an expensive one. To avoid expense, either the side entries should be driven on a pitch sufficient to allow the trains to run to gathering-up stations, nearer the working

i, or another system of haulage, such as main and tail
, or endless rope, should be adopted to reduce the cost of
and expensive local haulage.   Cases no doubt occur in
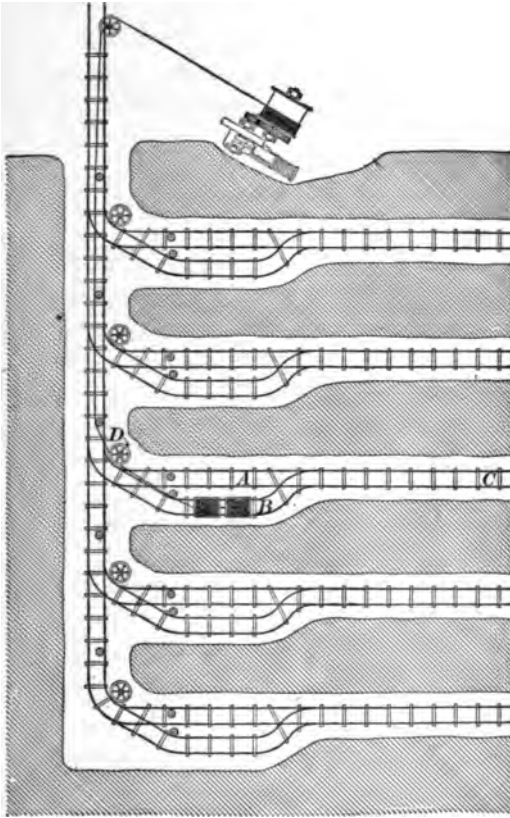
FIG. 842.

h the side entries are so short that the main and local
ige here shown would do, but they are exceptional.

147. In engine-plane haulage, it is important that
s of reasonable length be run; otherwise a greater num-
.f cars are required than it is prudent to use for the out-
   The number of cars in a train should not exceed
y, and the grade should not be less than 3 per cent. to

attain an average speed of 10 miles an hour when runnir back. A train of empty cars in good working order, a٦ running on a good track, will acquire a good speed on a pit٠ of 2.25 per cent., and a train of full cars under the sar conditions will run at a good speed on a pitch of 2 per cen· but for all-around good work, a pitch of 3 per cent. is t most reliable, and, therefore, should be the minimum.

CALCULATIONS FOR ENGINE-PLANE HAULAGE.

2348. To find the tension in the haulage-rope whe the inclination of the road, the length and weight of th rope, the number and weight of the cars, and the coefficien of traction are given, proceed as follows: First, find th٠ traction due to the friction of the weights of the cars, coal and rope as follows: Divide the sum of the weights of th٠ cars, coal, and rope by 40, the latter number being the co efficient of traction due to friction. Second, to find th٠ traction required for the gravity due to the incline, multi· ply the sum of the weights of the car, coal, and rope by the per cent. of grade, and the product is the traction re· quired for gravity. To find the total force required for traction, or the tension in the hauling-rope, add the traction due to friction to the traction due to gravity, and the sum is the tension in the hauling-rope. What has been here said can be shown by a formula.

Let W = the total weight of the train;
w = the weight of the rope;
$C = \frac{1}{40}$ = the coefficient of friction;
a = the grade, per cent.;
T = the tension in the rope in pounds.

Then, $$T = \left(\frac{W + w}{40}\right) + a\,(W + w). \qquad \textbf{(200.)}$$

EXAMPLE.— 20 loaded cars weigh 4,000 pounds each, and the hauling rope is 5,000 feet long and weighs .88 pound per foot. What is th tension in the rope at the moment the engine hauls away from th bottom of the incline, the grade being 3 per cent.?

SOLUTION.—The tension due to friction is equal to

$$\frac{(20 \times 4,000) + (5,000 \times .88)}{40} = 2,110 \text{ lb.}$$

The tension due to gravity is equal to

$$.03 \left[(20 \times 4,000) + (5,000 \times .88) \right] = 2,532 \text{ lb.};$$

and, therefore, the total tension in the rope is

$$2,110 + 2,532 = 4,642 \text{ lb.} \quad \text{Ans.}$$

The total tension can be found by substituting the values in formula **200**; thus,

$$T = \frac{(W + w)}{40} + a(W + w) =$$

$$\frac{(20 \times 4,000) + (5,000 \times .88)}{40} + .03 \left[(20 \times 4,000) + (5,000 \times .88) \right] = 4,642 \text{ lb.}$$

EXAMPLE.—Suppose that the train in the previous example is made to run with a velocity of 12 miles an hour. What would be the horse-power required to do this work?

SOLUTION.—The velocity of the train is $\frac{12 \times 5,280}{60} = 1,056$ feet per minute. The tension in the rope was found to be 4,642 lb. Hence, f 1,056 be multiplied by 4,642, the product will be the number of foot-ounds of work per minute the engine must do, and if this product be ivided by 33,000, the quotient will be the horsepower required.

Thus, $\text{H. P.} = \frac{1,056 \times 4,642}{33,000} = 148.5 \text{ H. P.} \quad \text{Ans.}$

2349. There is one peculiarity in the solutions that have ıst been arrived at, and that is the taking of the full weight ᶠ the rope. The student will observe that the engine must ıly overcome the total weight of the rope at the moment ᶠ starting the run, and at the finish of the run the weight ᶠ the rope has no effect; therefore, the mean weight of the ᵓpe against the engine is half the total weight. It would, ıerefore, appear that the total weight of the rope should ot be taken; but it so happens that as the weight of the ᵓpe reduces, the leverage against the engine increases. he engine begins to haul with an empty drum, and as the ᵓpe rolls on, the radius of the drum increases, and, there-re, if the engine runs at a constant speed, the speed of the ain quickens as the rope shortens. For this reason, the ᵣrect average of resistance is found by taking the total

weight of the rope throughout the run as an offset to the increasing radius of the drum.

EXAMPLE.— 25 loaded cars weigh 4,600 pounds each, the length of the engine-plane is 6,000 feet, the weight of the rope per foot is 1.2 pounds, the grade of the incline is 5 per cent., and the velocity of the train is 13 miles per hour. What is the tension in the rope and the required horsepower of the engine?

SOLUTION.— $W = 25 \times 4,600 = 115,000$ lb.;
$$w = 1.2 \times 6,000 = 7,200 \text{ lb.}$$

Substituting these values in formula **200**, we have

$$T = \frac{(W + w)}{40} + a(W + w) =$$

$$\frac{115,000 + 7,200}{40} + .05(115,000 + 7,200) = 9,165 \text{ lb.}   \text{Ans.}$$

The velocity of the train is $\frac{5,280 \times 13}{60} = 1,144$ ft. per min.; therefore, the horsepower is $\frac{9,165 \times 1,144}{33,000} = 317.7$ H. P.   Ans.

2350. The weight of the rope is such an important factor in the loss of useful effect on an engine-plane, that if the work is run off at a high velocity with a rope of light weight, it can be done with a less expenditure of energy. To prove this, suppose the same amount of work must be run off as in the previous example, but at a rate of double the speed and with a rope of one-fourth the weight; what horsepower is required to do the work with the lighter rope?

One-half of W in the preceding example is $\frac{115,000}{2} =$ 57,500. One-fourth of w is $\frac{7,200}{4} = 1,800$ lb., and the velocity per minute is, for 26 miles an hour, equal to $\frac{26 \times 5,280}{60} = 2,288$ ft. Using formula **200**, we have

$$T = \frac{(W + w)}{40} + a(W + w) =$$

$$\frac{57,500 + 1,800}{40} + .05(57,500 + 1,800) = 4,447.5 \text{ lb.,}$$

and $\frac{2,288 \times 4,447.5}{33,000} = 308.4$ H. P.   Ans.

From this calculation, it is plain that the loss of useful effect due to the heavy rope is equal to $317.7 - 308.4 = 9.3$ H. P.

2351. Engine-plane haulage, like all other systems, is capable of being modified for special conditions, and sometimes these modifications are of great importance. For example, the modifications may be such as to closely approximate to some of the modes of action of a main and tail rope haulage. In such a case, the haulage from four or more districts in a largè mine is done with separate ropes, that are made attachable and detachable with coupling-sockets. The haulage will be done with what are practically tail-ropes, because they are used for hauling from the engine instead of to it, the shafts being situated in a shallow basin of such a character that the loaded trains will run by gravity to the shaft, but have not sufficient fall to haul the empty trains away into the different stations in the workings. Each district rope, therefore, takes the exact form of a tail-rope, for if the cars are hauled into four districts A, B, C, and D, at each of the stations there is fixed a return wheel for the district tail-rope. To haul into any one of the districts, the method of coupling is as follows: One end of the rope is coupled with a socket to the rope on the drum, and the other end is coupled onto the inner end of the empty train for hauling in. The engineer then receives the signal, " Haul into A station," and when the train arrives the rope is knocked off and attached to a full train, and then the signal is given to the engineer, " Drop away the full train from A station."

This is a cheap and efficient system, where the conditions are such as have been stated.

2352. The district rope system of engine-plane haulage is also adopted in cases where the seam and the workings advance up grade from one side of the shaft and down grade from the other, and the seam is pitching sufficiently for the empty train to fall one way and for the full train to fall the other. The power required to do the work of this variety

twice the weight of one side of the rope is taken to
resistance due to friction. Second, the only resist
to gravity is that of the train of empty cars. The :
example shows how the horsepower is found for a
of this character:

EXAMPLE.—What horsepower is required to haul 30 empty
an incline 4,000 feet long, having a grade of 3 per cent.?
car weighs 1,400 pounds, the weight of the rope per foot o
.88 pound, and the maximum velocity of the train is 12 mile

SOLUTION.—The weight of the rope is $4,000 \times 2 \times .88 =$
and the weight of 30 empty cars is equal to $1,400 \times 30 =$
$\dfrac{(W + w)}{40} = \dfrac{(42,000 + 7,040)}{40} = 1,226$ lb., the resistance due t
and $a W = .03 \times 42,000 = 1,260$ lb., the resistance due to gravi
$1,226 + 1,260 = 2,486$ lb., the tension in the rope. Again, 1
hour is equal to $\dfrac{5,280 \times 12}{60} = 1,056$ feet per minute; ther
required horsepower is $\dfrac{1,056 \times 2,486}{33,000} = 79.552$ H. P. **Ans.**

2353. From the foregoing, the following equat
the tension in the rope of an engine-plane having
rope:

$$T = \frac{W + w}{40} + a\,W. \qquad \textbf{(201.)}$$

Also, if $v =$ velocity in feet per minute, and $H =$ th
power, the following equation gives the horsepower
to operate the plane:

TAIL-ROPE SYSTEM.

DESCRIPTION OF THE SYSTEM.

2354. There are four classes of roads on which **this** ystem of haulage may be adopted with success, namely:

Level roads.

Undulating roads.

Roads of small pitch.

Roads on which alternate levels and relatively high pitches occur.

2355. The mode of action that characterizes this ystem is that of a special provision for hauling in opposite irections. This is secured by means of two ropes, called, espectively, the main and tail ropes. Another feature of iis system is its adaptability for hauling trains of cars to nd from all the gathering-up stations of the different dis-iets in a mine. In hauling to the hoisting-shaft, it is clear iat the destination is the same for every loaded train, from hatever gathering-up station it may come; and as the uling-engine is either located in the neighborhood of the aft, or the hauling-ropes enter the mine through the shaft, is clear that the rope that is always pulled in one direction wards the shaft will have some specific name, and this me is *main rope;* that is, it is the one rope that does e hauling out of every district to the shaft. Hence its me, *main,* or chief rope, or rope that is always used for uling to one point. The same can not be said of any of : tail-ropes, for they are used for hauling to different sta-ns. To realize what their use is, suppose that there are : gathering-up stations, *A, B, C, D,* and *E*; then, to haul the *A* station, a special district tail-rope is required, for : rope must pull in the direction of the *A* station only, or station to which the train must go. This being so, the -rope for *A* will not do for *B*, neither will the *B* rope do the *A* station. The same is true of the other three sta-is. Therefore, if it is intended to haul out of five dis-ts, five tail-ropes, or one for each station, are required.

24—28

FIG. 843.

FIG 844.

The tail-ropes, then, are peculiar to the districts for whose haulage they are used.

2356. A very good idea of the use of the tail-rope may be obtained from a study of Fig. 843. In this case, a train of cars is supposed to be running on a level road hauled by the engine *A* to the shaft. It is clear that without a reverse or tail rope, this engine could not be applied to haul an empty train back to *B*; therefore, in the absence of a tail-rope engine, the engine *B* must do the return hauling. It will readily be seen that the rope running onto the drum of the engine *A* takes the place of the main rope, and the rope running onto the drum of the engine *B* takes the place of the tail-rope; yet, it is not truly a tail-rope, for the one is as much a main rope as the other.

The chief lesson this figure teaches is the difference between this system of haulage and that of the engine-plane. In the latter, gravity did the work that is done in the former by the tail-rope. As on a level plane there is no force like the earth's gravity to return the train to the point whence it is hauled by the engine *A*, the engine *B* is made to do the tail-rope or return haulage.

2357. Fig. 844 shows how the return haulage is done with a tail-rope. It will be noticed that the two drums on the engine at the left-hand side of the figure are for winding the two haulage-ropes. For example, *c* is the drum for the main rope and *d* is the drum for the tail-rope. The main rope is coupled to the front of the train of cars, and is seen to be hauling them to the shaft. The tail-rope, on the other hand, is uncoiling from its drum and passing along the side of the track on rollers at *a*, *a*, *a*, and ultimately it is seen passing around the return sheave at *S* to the rear end of the train to which it is attached.

Fig. 845 is an illustration of how the return sheaves are erected at the gathering-up stations at the ends of the haulage districts.

2358. Having so far given a general description of this system of haulage, it next becomes important to notice the

use and relationship of certain mechanical details with which the student must be familiar before he can claim to have an intimate acquaintance with this system of haulage. For example, it is not enough that he should know that two hauling-drums are used, but he should thoroughly understand the special work for which each is intended. It is not difficult to conclude that one of them is for the coiling and uncoiling of the main rope, and the other to do the same for the tail-rope, for it so happens that while one coils on, the other uncoils. For example, when the main rope is coiling on, the engine is engaged in hauling coal to the shaft from one of the stations, and, therefore, as the main rope is coiling on, the tail-rope must be uncoiling. As the train is approaching the shaft, it is hauling in its rear the tail-rope; when the train has reached the bottom of the hoisting-shaft, the tail-rope is uncoupled from its rear and attached to the front of an ingoing empty train, and at the same time the main haulage-rope is uncoupled from the loaded train and coupled to the rear of what is now the ingoing empty train. To effect a change in the direction of haulage, the tail-rope drum is thrown into gear, the main-rope drum is thrown out of gear, and the engine hauls in the empty train with the tail-rope. The empty train now pulls in the main rope to do the work of hauling out the next loaded train, just as the loaded train pulled out the tail-rope. It has just been stated that the drum for the main rope is uncoupled from the engine, and the tail-rope drum is coupled to the engine to haul in the empty train. This statement suggests some mechanical arrangement for connecting and disconnecting the hauling-drums with the engines. This operation is technically known as clutching in and out of gear. For instance, to haul in, the tail-rope drum is clutched onto the engine, and the main-rope drum is thrown out of gear. To haul the cars out, the drum of the main rope is clutched onto the engine, and the tail-rope drum is thrown out of gear. It might be thought that clutching one drum and throwing the other out of gear is all that the engineer must do. Such, however, is a mistake,

for, when passing over certain inequalities in the road in hauling in or hauling out, the engineer must put the brake on the drum that is out of gear, to keep the rope reasonably tight, and prevent the possibility of it uncoiling and kinking, for it is destructive to a rope to allow it to uncoil itself.

2359. So far, then, as the drums are concerned, the matter is clear enough, but the coupling of the engines to the drums is a matter that requires more than passing attention. There are two modes of gearing up the engines for hauling. In some cases, the hauling is done with second-motion engines; that is, the drum and the engines are on different shafts. In such a case, the engines are said to be on the second motion, the engine and the drums being geared so that the engine makes two or more revolutions to one revolution of the drum. This permits smaller engines to be used than when the engines and the drums are on the same shaft, in which case the engines are on the first motion. If the same amount of work is to be done in each of the two cases, the engines on second motion must be run at a higher speed than the engines on first motion. The latter are made larger in size, and consume the same amount of steam in a given time that the small engines do while running at a higher speed. Now, it might be thought that the question of engines on the first and second motions is not of much importance; but such is a mistake, for in large and extensive mines it is important that the hauling-engines be put on the first motion, where the roads will permit, as they are required to run heavy trains with despatch, and make up, for the extra length of road, for the time occupied in socketing and unsocketing the tail-ropes and for unpreventable delays at the different stations.

2360. The next matter that must be considered is that of the tail-ropes for the different districts, so that the methods of socketing and unsocketing them with the general tail-rope of the engine may be understood. To make the explanation clear, Fig. 846 is introduced. Here the general haulage of the mine is divided into five gathering-up stations;

namely, A, B, C, D, and E. In the diagram a loaded train
is seen to be leaving the gathering-up station B on its way
to the hoisting-shaft. The drum for the main rope is shown
at R, and the main rope is seen along the middle of the road.
From the tail-rope drum marked t, the tail-rope is seen to
advance along the lower side of the principal haulage-road,
enter the entry B by the deflecting sheaves e, and then pass
up along the right-hand side of the entry to the return
wheel w, around which it passes, and returns to the rear of
the train to which it is attached. On looking at district A,
it is seen that both sides of the tail-rope from the return
wheel reach to the entrance of the station, where the two
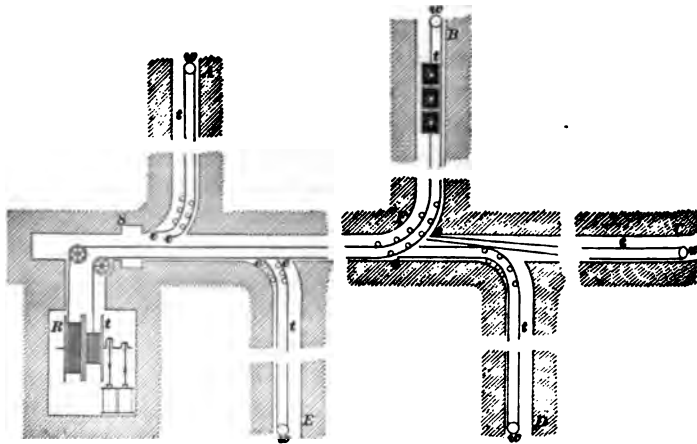
FIG. 846.

ends are seen lying ready for attachment. The same appears
in districts C and D. On looking thoughtfully at these tail-
ropes, a moment's reflection will enable the student to un-
derstand that if the train is hauled out from the shaft to F,
it can be disconnected from the B tail-rope, and connected
with the C one by the socket at the off-take station, and by
this means hauled into station C. In this way, disconnec-
tions and connections can be made for running into any of
the other stations. It is plain that to haul into any district,
a change must be made in the tail-rope sockets at the en-
trance to it.

2361. There are two methods in practice for socketing and unsocketing the tail-ropes. In the first and oldest one, the connections are made when the train arrives at the entrance to the district into which it is intended to go. In the second method, all the district connections are broken continuously up to the entrance of the district for which the train is destined. For example, suppose the last train arrived out of district *A*. The coupler for station *A* uncouples his district tail-rope and couples up the general tail-rope to run past his station; the other couplers do the same, except the one who has signaled for the next train. He uncouples the general tail-rope, couples up the tail-rope of his district, and then signals for the engineer to "run in," and the train runs continuously from the shaft into the required station without a stop. This system of socketing is decidedly the best, because it prevents all unnecessary delay, and never produces a hitch when an efficient code of signals is adopted. Again, the system secures great economy in steam, for the engine is kept running almost continuously.

2362. Figs. 847, 848, and 849 show different methods of socketing. In cases like these, a main or general tail rope is used for all the entries. The connections, in the case of
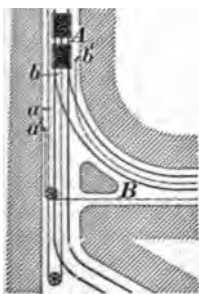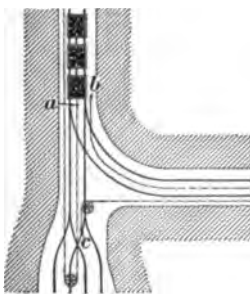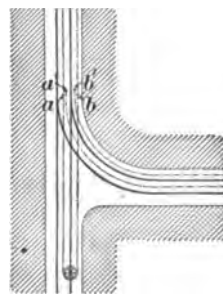
FIG. 847.   FIG. 848.   FIG. 849.

Fig. 847, are made when the empty cars come in to *A*. The rider unsockets the lead rope at *b*, and sockets on the entry-rope at *b'*. At the same time he breaks the connection at *a*, and couples on the rope *a'*. He then signals to the engineer to haul into the required gathering-up station.

The pulling out of the loaded cars is done by simply reve rs. ing the operation.

2363. In Fig. 848 another method of socketing is adopted; but this can only be put in practice in exceptional cases. The intention of this arrangement is to reduce the number of sockets. The coupling of the parting tail-rope to the main rope is made at *c*. The main rope is also uncoupled at *a*, and the rope *b* is coupled onto the train. Then the train can be hauled into the gathering-up station in the parting. Fig. 849 illustrates the branch connections for coupling-up, so as to run right through from the shaft into the station. The operation of uncoupling and coupling is done at one place. The ropes *a* and *b* are uncoupled from the train coming in the main road, and ropes *a'* and *b'* are coupled on. That portion of the main-roadway rope lying between the coupling-point and the sheaves lies idle in and alongside the road.

CALCULATIONS FOR TAIL-ROPE HAULAGE.

2364. The calculations for the tension in the rope and the horsepowers required for main and tail rope haulage require modifications that are not applied in other systems of haulage. For example, instead of gravity increasing the tension in all cases, it sometimes decreases it, and, therefore, becomes a negative quantity. To make all these differences clear, however, the conditions occurring will be explained in every case. The meaning and use of the letters in the expressions are as follows:

W = the total weight of the loaded train;
W_t = the total weight of the empty train;
w = the weight of the rope;
$C = \frac{1}{40}$ = the coefficient of friction;
a = the grade per cent.;
T = the tension in the main rope in pounds;
T_1 = the tension in the tail-rope in pounds.

2365. The weights per linear foot of the main and tail ropes are sometimes different; sometimes the main rope is the heavier, and at other times the tail-rope is the heavier.

In all cases, however, the difference is small, and for that reason the weight will be taken as the same in the following examples, unless otherwise stated. Again, when an empty train leaves the hoisting-shaft, the weight of the moving tail-rope is equal to twice that due to the distance from the shaft to the gathering-up station, and when the train reaches the gathering-up station, the weight of the rope moving is still equal to twice the length of the journey, because then one length of the main rope lies in the middle of the track, and an equal length of the tail-rope lies on the side of the track. Therefore, the weight of rope in motion is never more or never less than that due to twice the length of the track from the shaft to the terminus to which the train must be hauled. For this reason, the weight of rope due to twice the length of the track will be taken in each case.

2366. The tensions in the main and tail ropes are calculated on the longest runs and on the maximum up grades of the roads. In cases where the haul to the shaft is down grade, and, conversely, the haul to the gathering-up station is up grade, the greatest tension falls on the tail-rope.

When the ropes weigh the same per linear foot, the tension in the main rope may be found by formula **201.** The tension in the tail-rope is found by the same formula, except that the weight of the empty train is to be used. That is, using the notation of Art. **2364,**

$$T_1 = \frac{W_1 + w}{40} + a\,W_1.$$

EXAMPLE.—The greatest length of main and tail rope haulage in a certain mine is 6,500. feet, and the tracks are perfectly level. The weight per foot of the main rope is .7 pound, the weight per foot of the tail-rope is .6 pound, the full cars weigh 5,000 pounds, the empty cars weigh 1,800 pounds, and the trains consist of 20 cars. What are the tensions in the main and tail ropes? If the average speed of the trains is 10 miles an hour, what is the horsepower of the hauling-engine, due to the maximum tension of the ropes?

SOLUTION.—The weight of the train of loaded cars is 20 × 5,000 = 100,000 lb. The combined weight of the two ropes is (.7 + .6) × 6,500 = 8,450 lb. The tension in the main rope is

$$T = \frac{(W + w)}{40} = \frac{(100,000 + 8,450)}{40} = 2,711.25 \text{ lb.} \quad \text{Ans.}$$

The weight of the train of empty cars is equal to $1,800 \times 20 = 36,000$ lb. The joint weight of the ropes is $(.6 + .7) \times 6,500 = 8,450$ lb., as before.

Then, $T = \dfrac{(W_1 + w)}{40} = \dfrac{36,000 + 8,450}{40} = 1,111.25$ lb., the tension in the tail-rope. Ans.

According to the conditions of the example, the horsepower must be calculated from the maximum tension. The speed of the train is 10 miles an hour, or $\dfrac{5,280 \times 10}{60} = 880$ feet per minute. Then, applying formula **202**,

$$H = \frac{Tv}{33,000} = \frac{2,711.25 \times 880}{33,000} = 72.3 \text{ H. P.} \quad \text{Ans.}$$

2367. It sometimes occurs that a portion of a haulage-road is up grade, and there the maximum strain on the rope takes place.

EXAMPLE.—On a short portion of a main and tail rope haulage the main rope must haul a train of 30 loaded cars up a grade of 4 per cent. What is the maximum tension in the main rope when a full car weighs 5,000 pounds, the main rope weighs 1.2 pounds per foot, the tail-rope weighs .88 pound per foot, and the length of the track is 5,600 feet?

SOLUTION.—The weight W of the train is $5,000 \times 30 = 150,000$ lb., and the weight w on the ropes is $(1.2 + .88) 5,600 = 11,648$ lb. Then, applying formula **201**,

$$T = \frac{(W + w)}{40} + aW = \frac{(150,000 + 11,648)}{40} + .04 \times 150,000 = 10,041.2 \text{ lb.},$$

the maximum tension in the rope under the given conditions. Ans.

2368. When a haulage-road only runs up grade for a short distance, it is the practice to accelerate the speed of the train a little before reaching the rising ground, and then by its inertia the train is carried over with no other loss than that of a reduced velocity, which it soon recovers. The horsepower is, however, increased for this short length of up-grade work in the following proportion:

Let l = the full length of the road;

     l_1 = the short length of the up-grade road;

     P = the horsepower required for level track;

     P_1 = the increased horsepower.

Then, $P_1 = \dfrac{P(l + l_1)}{l}.$ **(203.)**

The increased tension in the rope due to the local
-grade inclination of the road will very seldom equal
: tension due to friction alone; therefore, the expression
/en in formula **203** is sufficient; but where the inclination
so great as to require a higher tension than that due to
ction, then the increase of power is found by a special
lculation.

EXAMPLE.—A main and tail rope haulage-road is for five-sixths of
length level, but in hauling out to the shaft, ¼ of the length of the
d is up grade.   If the road was all level, the haulage could be done
h an 80-horsepower engine.   What should be the power of the
;ine according to formula **203**?

SOLUTION.—Substituting in formula **203**, we have

$$P_1 = \frac{P(l + l_1)}{l} = \frac{80 \times (1 + \frac{1}{5})}{1} = 93\frac{1}{3} \text{ H. P.}\quad\text{Ans.}$$

n this connection, it must be understood that the up
.de encountered is but slight, and that the speed of the
in on the level is accelerated so as to carry the train along
inertia.   In any case, the formula gives only approxi-
te results.

369.   EXAMPLE.—In a main and tail rope haulage in a certain
e all the roads leading to the shaft have a mean fall of 3 per cent.,
greatest length of run is equal to 4,862 feet, and the mean velocity
3 miles an hour.   The hauling-ropes weigh .88 pound per foot,
trains consist of 25 cars, each loaded car weighs 5,000 pounds, and
mpty car weighs 1,900 pounds.   What is the tension in the main
tail ropes, respectively, and what is the required horsepower
he hauling-engine?

SOLUTION.—To find the tension in the main rope, the gravity force
to the pitch of the incline must be treated negatively, because it
ices the tension in the rope; then, formula **201** becomes

$$T = \frac{W + w}{40} - a W.$$

ince $W = 5,000 \times 25 = 125,000$ lb., $w = 4,862 \times 2 \times .88 = 8,557.12$ lb.,
$\frac{3}{100} = .03$, we have

$$T = \frac{W + w}{40} - a W = \frac{(125,000 + 8,557.12)}{40} - (.03 \times 125,000) =$$

11.072 lb., the negative tension in the main rope.   Ans.

'his means that not only is there no tension in the main rope, but
xcess of gravity force equal to 411.072 lb., which will, without the
ine, run the train at a high velocity.

The gravity force in the case of hauling the train of empty cars is positive, because they are made to ascend the incline; hence, we can apply formula **201** directly to find the tension in the tail-rope.

$W_1 = 1,900 \times 25 = 47,500$ lb.; $w = 4,862 \times 2 \times .88 = 8,557.12$ lb., and $a = \frac{1}{33} = .03$. Therefore,

$$T_1 = \frac{W_1 - w}{40} + a\,W_1 = \frac{(47,500 + 8,557.12)}{40} + .03 \times 47,500 = 2,826.428 \text{ lb},$$

the tension in the tail-rope. Ans.

No horsepower is exerted through the medium of the main rope, because the tension is negative; but by using formula **202**, the horsepower exerted through the tail-rope can be found. As the velocity is equal to $\frac{5,280 \times 12}{60} = 1,056$ feet per minute, we have

$$H = \frac{Tv}{33,000} = \frac{2,826.428 \times 1,056}{33,000} = 90.44 \text{ H. P.} \quad \text{Ans.}$$

2370. Let us observe the importance of calculating the tensions in the main and tail ropes. It is evident that if the main rope is a very light one, not only will the engine be better balanced, but great economy will accrue from the use of a less costly rope. In a case like this, however, it is better to reduce to a minimum the size of the main rope, and to increase the size and weight of the tail-rope to equalize the work of the engine. This conclusion will be clearly evident in the next example.

Where the haulage-roads have a fall towards the bottom of the shaft, it is evident that the full cars will furnish a gravity force that is negative to the general resistance, and that the difference between the weight of the ascending heavy tail-rope and the descending light main rope will furnish a gravity force that is positive to the general resistance.

Under these conditions, the tension in the main rope becomes

$$T = \frac{W + w}{40} - a\,(W - w_1), \qquad (204.)$$

where $w_1 =$ the difference in the weights of the ropes, and the other letters represent the same values as given to them in Art. **2364.**

To find the tension in the tail-rope, the same formula is used, except that the second term must be added.   That is,

$$T_1 = \frac{W_1 + w}{40} + a\,(W_1 - w_1).   \qquad \textbf{(205.)}$$

EXAMPLE.—If in the example of Art. **2369**, the main rope and tail-rope weigh .6 pound and 3.65 pounds per foot, respectively, what will be the tension in each rope, and what will be the required horse-power of the haulage-engine ?

SOLUTION.—Using formula **204**, we have

$$T = \frac{W + w}{40} - a\,(W - w_1) = \frac{5,000 \times 25 + 4,862\,(3.65 + .6)}{40} - .03$$

$$[125,000 - 4,862\,(3.65 - .6)] = 336.46\text{ lb.} =$$

the tension in the main rope.   Ans.

To find the tension in the tail-rope use formula **205**.   In this case, $W_1 = 1,900 \times 25 = 47,500$; hence,

$$T_1 = \frac{47,500 + 20,663.5}{40} + .03\,(47,500 - 14,829.1) = 2,684.2\text{ lb.}   \text{ Ans.}$$

Using formula **202**, we have

$$H = \frac{Tv}{33,000} = \frac{336.46 \times 1,056}{33,000} = 10.8\text{ H. P.,}$$

the horsepower required for the haul out with the main rope.   Ans.

Again, $$H = \frac{Tv}{33,000} = \frac{2,684.2 \times 1,056}{33,000} = 85.9\text{ H. P.,}$$

the required horsepower for the tail-rope haulage.   Ans.

2371. From the example just worked out, some interesting lessons are learned.   The first is, that the tension in the ropes under the greatest stress does not much exceed one long ton; further, the weight of a rope and the weight of a train may become negative to the resistance of friction. Again, the energy that is lost, when a portion of the gravity force due to the inclination of the road is wasted by using a brake, may be utilized by adjusting the weights of the ropes to retain it.

In the example of Art. **2369**, the horsepower was found to be 90.44, and in the example in Art. **2370** it is 85.9. Therefore, $90.44 - 85.9 = 4.54$ horsepower is saved by increasing the weight of the tail-rope.   More than this could be saved by further reducing the weight of the main rope, and further increasing the weight of the tail-rope, where, as in this case, there is a down grade to the shaft.

2372. In main and tail rope haulage it is important that the maximum velocity of the trains should never exceed 12 miles an hour, for three very important reasons: (1) the tracks are costly to keep in repairs when the velocities exceed the limit just given; (2) high speeds are destructive to the cars, and, therefore, increase the haulage cost per ton; (3) derailing is more frequent at high speeds than low ones. As the maximum speed must be kept within a safe working limit, the number of cars in a train is determined by five important factors, all of which are variable, or different in different mines. Before estimating the number of cars that should be attached in a train, the values of the factors just referred to must be known. They are:

1. The output of the mine in tons of coal per day.
2. The mean lengths of the roads.
3. The weight of coal a car will carry.
4. The number of trains that can be run out of the different districts.
5. The number of tons of coal each haulage district can produce.

2373. The values of the factors are found as follows:

The prospective output of the mine is found by estimating, on the basis of present practice, how much more is possible.

The lengths of the roads are calculated prospectively from the attainable lengths measured on the map of the available field.

The cars are made to carry such weights as the dimensions of the hoisting-shafts and the heights of the haulage-roads will allow.

The number of trains is found as follows:

(*a*) Find the mean of all the lengths of the districts, each being measured from the shaft to the making-up stations.

(*b*) Multiply the mean length of the districts by 3, for the following reasons: A journey of full cars out and empty cars in is equal to double the length of a district road, and to compensate for unpreventable and unforeseen delays, another addition must be made to the length of the track,

and, therefore, the mean length of the district roads must be multiplied by 3.

(*c*) Find the number of feet a point in the rope will pass through in one working day. Suppose, for example, the mean speed of the rope is 12 miles an hour, and that the time of one day is 10 hours; then, $5,280 \times 12 \times 10 = 633,600$ feet, the distance a point in the rope will move through in 10 hours.

(*d*) Divide the distance a point in the rope would move through if kept continually in motion by 3 times the mean length of the district roads, and the quotient will be the number of trains that can be hauled out per day.

EXAMPLE.—How many trains can be run out by a main and tail rope haulage in one day of 10 hours, the speed of the rope being 12 miles an hour, and the lengths of five districts being as follows:

$$A = 5,012 \text{ feet};$$
$$B = 4,654 \text{ feet};$$
$$C = 3,278 \text{ feet};$$
$$D = 7,101 \text{ feet};$$
$$E = 2,794 \text{ feet}.$$

SOLUTION.— $5,012 + 4,654 + 3,278 + 7,101 + 2,794 = 22,839.$

$22,839 \div 5 = 4,567.8$, the mean length.

Then, $\dfrac{5,280 \times 12 \times 10}{4,567.8 \times 3} = 46.23$, or, practically, 47 trains per day.   **Ans.**

To find the number of cars in a train, *divide the output in tons per day by the number of trains, multiplied by the tons of coal a car will carry; the quotient will be the number of cars in a train.*

EXAMPLE.—The output of a mine is 2,000 tons of coal per day; the number of trains to haul out this quantity is 47. If one car carries 2 tons, how many cars must there be in a train to do the work?

SOLUTION.— $\dfrac{2,000}{47 \times 2} = 21.276$, or, as there can not be a fraction, the number is 22 cars in a train; or, combining the examples,

$$\frac{2,000}{2} \div \frac{5,280 \times 12 \times 10}{4,567.8 \times 3} = 21.63, \text{ or } 22, \text{ nearly, as before.}   \textbf{Ans.}$$

EXAMPLE.—The following particulars are required for the construction of a haulage plant on the principles of main and tail rope:

(*a*) The number of trains that can be run out per day.

(*b*) The number of cars in a train.

(*c*) The horsepower of the haulage-engine.

The calculations must be based on the following particulars:

1. Six district haulage-roads, A, B, C, D, E, and F the lengths which are to be as follows:

$$A = 6,784 \text{ feet long;}$$
$$B = 4,250 \text{ feet long;}$$
$$C = 8,276 \text{ feet long;}$$
$$D = 3,560 \text{ feet long;}$$
$$E = 5,720 \text{ feet long;}$$
$$F = 7,963 \text{ feet long.}$$

2. The mean up grade to the shaft is 2 per cent.
3. The output is 3,000 long tons of coal in 10 hours.
4. The cars each carry 2½ tons.
5. The speed of the train is 10 miles an hour.
6. An empty car weighs 1,900 pounds.
7. The weight of one foot of the rope is 1.56 pounds.

SOLUTION.—(*a*) The distance a point in the main haulage-rope will run in 10 hours is $5,280 \times 10 \times 10 = 528,000$ ft.

To find the number of trains that can be run out in a day, this last product is divided by three times the mean length of the haulage-roads. The mean length is

$$(6,784 + 4,250 + 8,276 + 3,560 + 5,720 + 7,963) \div 6 = 6,092\tfrac{1}{3} \text{ ft.}$$

Hence, the number of trains is

$$\frac{528,000}{6,092\tfrac{1}{3} \times 3} = 28.89, \text{ or, practically, 29 trains. } \textbf{Ans.}$$

(*b*) To find the number of cars in a train, the output is divided by the number of trains multiplied by the number of tons a car will carry.

Thus, $\dfrac{3,000}{29 \times 2.5} = 41.4$ cars, or, practically, 42 in a train.   **Ans.**

(*c*) Remembering that the output is in long tons, the weight of a loaded car will be $(2,240 \times 2.5) + 1,900 = 7,500$ lb. Therefore,

$W = 7,500 \times 42 = 315,000$ lb., and $w = 1.56 \times 6,092.16 \times 2 = 19,007.54$ lb.

Substituting these values in formula **201**, we have

$$T = \frac{(W + w)}{40} + a\,W, \text{ or } \frac{(315,000 + 19,007.54)}{40} + .02 \times 315,000 =$$

$14,650.19$ lb.; the velocity of the trains is equal to $\dfrac{5,280 \times 10}{60} = 880$

feet per minute

Substituting in formula **202**, we have

$$H = \frac{T v}{33,000} = \frac{14,650.19 \times 880}{33,000} = 390.7 \text{ H. P. } \text{ Ans.}$$

TAIL-ROPE COUPLINGS.

2374. Fig. 850 shows a tail-rope coupling for connecting the different sections of the rope to run the train into a given district. There are many different coupling-links in use, and they all aim at securing three things:

First, to make a secure and reliable connec·tion.

Second, to provide a coupling-link that will knock as little as possible on the rollers, and not injure the coils of the rope on the hauling-drum.

Third, to furnish a coupling in which the connection can be made and unmade in as short a period of time as possible.

The coupling shown in Fig. 850 is one of the simplest and oldest in use, and it is here given to explain the general principle involved, but not as a model of a good socket. Sockets and couplings must be seen in use to be understood in their mode of action, but the following figures furnish a general idea for a student in pursuit of a knowledge of the important details. This form of coupling is even yet extensively used; it consists of two goose-neck fastenings *M* and *M* made in the form of a pair of trough-shaped tongs, riveted by three or four rivets to the end of the ropes, and connected by the links *N*, *N*, and *O*, as shown. The wires at the end of the ropes should be either welded or soldered for an inch or so to prevent untwisting, or otherwise the rope will draw out.

2375. The hauling-ropes for engine and gravity planes and for the main and tail rope planes have securely fastened on their ends, caps or sockets for coupling them up with the

In Fig. 855 is shown a knock-off link that can be operated by hand.   When a detachment is necessary, the lever *A* i
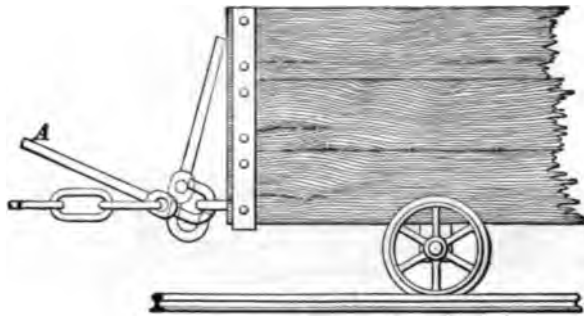
FIG. 855.

pulled upwards, and then the clevis, or hook, takes the position shown by the dotted lines, and the rope slips off.
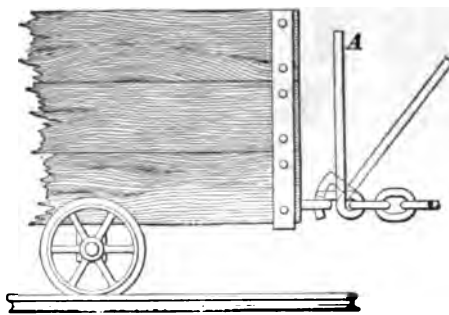
FIG. 856.

Fig. 856 is an illustration of a hand-detaching arrangement very similar in character to the previous one.

Fig. 857 is an illustration of an automatic detaching or disengaging hook. In this case, instead of the lever being operated by hand, one arm of a bell-crank lever is made to detach the rope by striking a beam *B*, as seen in the upper portion of the front end of the car.   The knock-off girder *B* is set across the track, and so fixed that a train passing under it causes a bell-crank *A* to strike against *B*, when the lever *C* disconnects the clevis and sets the rope and its coupling free.

Fig. 858 is another illustration of an automatic detachment, but, unlike the others, it does not break the connection until the engine stops.   The moment the car attempts to overrun the chain, the clevis detaches the rope, or rather the link *G*

slides off the hook at *H*, and the rope falls to the ground,
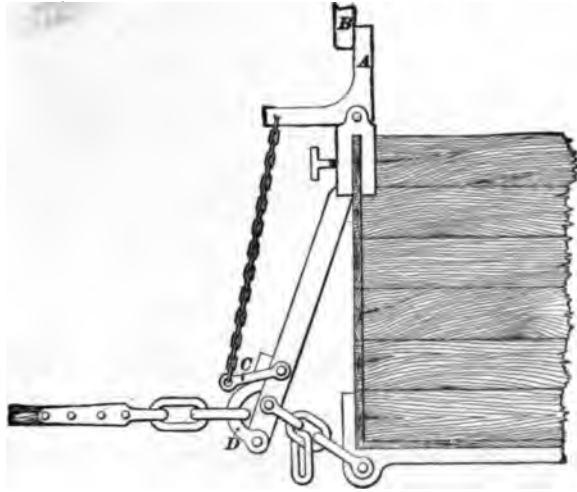
FIG. 857.

and leaves the train free to run farther on by its own momentum.

2377. The engines and drums for main and tail rope haulage are not always situated in the mines, for there the steam is either produced by boilers situated in the mines—which, to say the least, is a dangerous practice—or the steam is conducted in pipes from the surface. However well the pipes are protected, there is a great loss of energy due to the radiation of heat from the steam, and, therefore, it is better, if possible, to locate the hauling-engines at the surface, and conduct the hauling-ropes down the shaft or slopes into the mine. In some cases, the ropes are conducted through bore-holes; where this is done, great advantages are often secured, for sometimes by this method a large amount of local haulage can be effected that would otherwise have to be done by mules. At other times, it secures the advantage of a very much shorter rope. It might be thought that, by locating the engines at the surface and conducting the ropes through shafts or bore-holes, a difficulty would arise with

the signals between the gathering-up stations and the engineer, and between the gathering-up stations and the "make and break" boys at the entrances to the district roads Experience, however, has disapproved the conclusion, insomuch that the signals are found to be as perfectly given and as perfectly received as they would be if given from the gathering-up stations to the engineer in the mine. The necessity of accuracy in signaling can be readily appreciated when
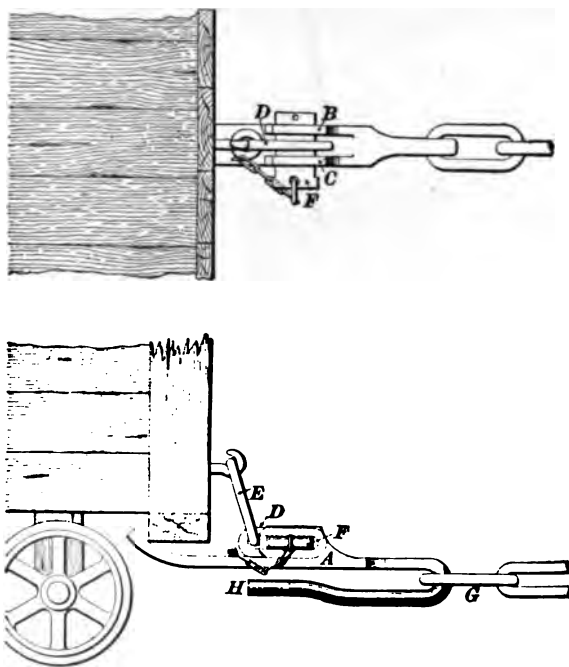
FIG. 858.

the student remembers that many of the roads are undulating, and require, on the part of the engineer, great care to prevent the trains overrunning the main rope on the one hand and the tail-rope on the other, when the trains are running down grade. With a proper code of signals, and with the engineer duly informed of the characteristic down grades of the several districts, the trains are kept well under control. In some cases, double tracks have been tried for main and

il rope haulage; but, so far, such a mode of proceeding has
een found to be neither prudent nor economical, as wide
ads must be expensively secured, and in many cases middle-
mbered.   This is expensive enough, but it is only a frac-
ion of a greater expense that is sure to arise in cases where
train running at a velocity of 10 or 12 miles an hour be-
omes derailed, and runs into the timber and draws it out.
'hen, if the roof is at all tender, it caves in, and, before the
illing stone can be removed and the roof can be resecured,
great loss arises from the stoppage of the work, in ad-
ition to the great expense of retimbering.   Single roads
re better for main and tail rope haulage, and the single-
ad system is what gives to the main and tail rope haulage
s preference over other systems.

ENDLESS-ROPE SYSTEM OF HAULAGE.

DESCRIPTION OF THE SYSTEM.

2378.   The endless-rope system of haulage can often
e substituted with advantage for either of the other three
rstems previously mentioned.   The underlying principle
 its action is that the haulage is done by a band or a series
 bands of rope that operate the cars like an elevator-chain
es the elevator-buckets.   To realize this, let the loaded
irs take the place of the full elevator-buckets, then the in-
rted and empty buckets are the exact analogue of the
npty cars; for on one side of the endless rope there are full
rs moving progressively to the shaft, and on the other the
ipty cars are moving inwards to the workings.   In another
nse, the principle of action of the elevator and the endless-
pe haulage is alike; that is, the buckets on the endless
ain of the elevator are set separately and at equal dis-
nces along the sides of the chain-belt, and in much the
me way the cars attached to the endless rope are set
parately at fixed distances along the rope, the full cars
ing attached to one side of the band of rope and the empty
es to the other.

2379. A good illustration of the arrangement of the rope-band, in reference to three of its most important features, is shown in Fig. 859.

1. The engine and the grooved wheels for clutching the rope for hauling are seen in plan in the upper portion of the figure, and in elevation in the lower portion. It will be noticed that the engine is located at one side of the tracks.

2. The two parts of the endless rope are seen between each of the two tracks, for the purpose of hauling in opposite directions, as shown by the arrows, which indicate that the loaded cars are moving outwards along the lower track to the shaft, while the empty cars are moving inwards to the workings, as indicated by the arrow on the upper track.

3. A tail-wheel C is provided to form the inner loop or end of the rope-band; hence, the two sides of the endless rope run along their respective tracks and around the tail-wheel C and the grooved wheels of the engine at A and B.

Two other important provisions designed to keep the rope tight are as follows: First, the tension-weight D. This is employed to keep the rope-band tight on the grooved grip-wheels; for if this is not done the rope will slip and the engine will be incapable of doing its work. Second, the tail or return wheel C is made, by means of a tension-balance, to keep the inner loop of the rope tight; for should one of the sides of the rope-band be allowed to run slack, then the grips attached to the cars lose their hold of the rope, and the haulage on that side of the band ceases. It will be seen that several deflecting sheaves are required for the engine end that are not necessary for the tail-sheave end. The reason for the difference is evident; for example, the sheave C can be made to slide in a frame inwards or outwards as the tension of the band requires adjustment, but the engine and the grip-wheels can not be made to slide in this manner; consequently, the tension-balance truck D has a wheel mounted on its top, and this forms a loop on one of the sides of the rope-band. Should the band slacken, the truck D descends the incline by its weight and tightens the rope, and should the rope become for a moment over-tight, then the tension-bal-

k *D* rises and releases the stress. The deflecting
;, *G*, *H*, *I*, etc., are necessary for the fixed engine,
it required for the tail-wheel *C*.
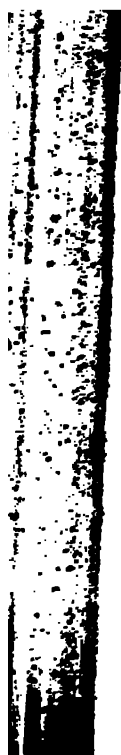
Fig. 860 is a plan of a typical endless-rope haul-
a single band. At the right-hand end of the figure
the tracks *S* and *s* begin at the hoisting-
shaft. The engine is located at *E*, and
the sides of the rope from the grip-wheels
are seen to pass round the deflecting
wheels *w*, *w*. The tail, or return, wheel
is shown at *T*. The direction of the
motion of the loaded cars *l*, *l*, *l*, etc., is
towards the hoisting-shaft, as indicated
by the arrows, and the direction of the
empty cars *c*, *c*, *c*, etc., is inwards to the
working face, as shown by the arrows.
In this figure, the similarity of the end-
less-rope haulage to that of the elevator
endless chain and buckets is plainly seen,
and yet there are three particulars in the
mode of action in which the endless-rope
haulage differs from the elevator chain and
buckets.

1. The buckets of the elevator are per-
manently fastened to the chain-belt;
whereas, in the case of the endless rope,
the cars are only temporarily attached.

2. In the case of the endless rope,
the cars are detached at the hoisting-shaft
end of the loop and passed on to the cage,
and the empty cars from the surface are at-
tached to the ingoing side of the rope-band.

3. At the tail-wheel end of the rope-
band the empty cars are detached and
sent into the workings to be refilled,
while loaded cars from the workings are
ly being attached to the outgoing side of the rope

FIG. 860.

to be carried forwards to the hoisting-shaft. The elevator chain and buckets are self-filling and self-emptying, while the cars of the endless-rope haulage are continually in process of being attached and detached. That is to say, at the tail-wheel or inner end the empty cars are continually being detached and the full cars are being attached, while at the outer or shaft end of the endless-rope band the full cars are always being detached and the empty cars are being attached.

2381. Fig. 860 shows an endless rope with a single band. Sometimes surface haulages of this character extend for miles over undulating ground. In the mines, however, it seldom occurs that a single band is sufficient for the entire haulage of a seam, and, therefore, several deflecting bands are necessary. When this is the case, the tail-wheel for the main or principal band is made to act as a grip or fleet wheel, and to rotate by suitable gearing another fleet-wheel, which hauls from the workings the cars coming to the main band. Therefore, the geared fleet-wheel for the local haulage is made so that it can be clutched in and out of action to prevent needless running when the supply of loaded cars from any district is not sufficient to keep this secondary band continually moving. Sometimes the main band is continued forwards in its advance into the workings, and deflecting grip-wheels are fixed at the entrances to all the districts it passes. When this arrangement is used, four or five deflecting bands bring their contents to one main band. The statement of this fact suggests that the cars on the main band must run at a higher velocity than those on the branching bands, and, in addition, that the cars on the district bands must be set at greater distances apart, because the main band must run off the cars from all the districts.

Since qualifications of this character are necessary, it is clear that the relative velocities of the secondary bands of rope must be properly adjusted, or the haulage, instead of being a success, will be a failure. Sufficient has been said to furnish a general description of the points which give to the endless-rope haulage its individuality.

2382. The conditions under which endless-rope haul-
ge can be applied in mines to secure special advantages
ıust next be considered.

1. A gravity-plane worked with an endless rope gives
ɛtter all-round results than a plane worked with two ropes
ıd two trains, because with the endless rope there is prac-
:ally no limit to the length of the haulage which can be
ıne by gravity. As was stated previously, when two
pes are used, the weight of the rope attached to the empty
ain at the commencement of the run soon becomes suf-
:ient to counterbalance the weight of the coal.

2. Ordinary gravity-planes can only be made efficiently
lf-acting on a down grade for the loaded cars, whereas an
ıdulating plane with endless rope that alternately pitches
opposite directions can be made self-acting by gravity, if
has a general fall sufficient to counteract the friction due
traction.

3. The united advantages of an endless-rope gravity-plane
e such that it works efficiently wherever the two-rope
stem answers, and it works and gives satisfactory results
ıere the two-rope system fails; that is, on very long planes
d on planes pitching in opposite directions, yet having a
fficient general fall.

4. The general haulage of a mine can be done by gravity
:ough the medium of an endless rope, where the ordinary
avity-plane could not be applied.

5. More work can be accomplished at a mine by the
dless-rope system than by an engine-plane, because the
·un back " on an engine-plane is done by gravity, and un-
s the pitch is sufficient to produce a high speed, only a
atively small amount of work is done, and where the
:ch is considerable, a large amount of work is wasted in
:ing a long, heavy rope.

6. In some cases, where an engine-plane is down grade to
ɛ shaft, and so long that the two-rope gravity-plane will
t act, and engine-power must be employed to assist
avity, the endless rope will act and give satisfactory results
:hout the aid of steam-power.

7. All the haulage that is done by the main and tail rope system can, where the roof and floor will permit, be done better and more cheaply by the endless-rope system, because the work can be run out of all the different districts the same as with the main and tail rope, yet with a smaller expenditure of motive power.

8. Haulage by the endless-rope system is cheaper on undulating roads than the main and tail rope, because the only work expended in the haulage is that of traction and the gravity of the mean pitch.

9. With the endless-rope haulage, there is no congestion at the gathering-up stations, and no delays due to several districts calling for cars at the same time; for the endless rope secures a continual output of full cars and a continual income of empty ones to keep the work progressing.

2383. So far, attention has been given to the typical endless-rope system. There are in practice several modifications of it, specially intended to adapt it to conditions in which the double track can not be used, as where the roof and floor are tender, and can not be kept secure without risk and great expense. These modifications aim at using a single track instead of a double one. In this case, the diffusion of the cars along the tracks must be dispensed with, and, in lieu of this, several cars are made to run in trains. For this arrangement to answer with an endless rope, two special provisions must be made. First, partings, or pass-bys, must be provided at frequent intervals along the track; second, automatic grips must be used, so that when a train enters a pass-by, the grip that secures it to the rope is automatically unloosened, and the train stands until a train moving in an opposite direction reaches the point in the track that is exactly opposite to that of the train in the parting. Then the grip of the standing train must automatically reclose, and allow it to move on to the succeeding pass-by. The pass-by and the automatic grip are only different substitutes for securing the advantages of the typical endless rope with double tracks. These modifications, how-

ot remove, but only reduce the danger and **expense**
ȝ and maintaining double tracks, for double tracks
at the partings. They are expensive to maintain
in good order, and experience has shown that the
ȝpe haulage with trains and numerous partings is
not an improvement on the main and tail rope sys-
is not, for an all-round haulage, equal to it.

, There are two distinct methods of arranging the
endless-rope haulage. In one of them that has
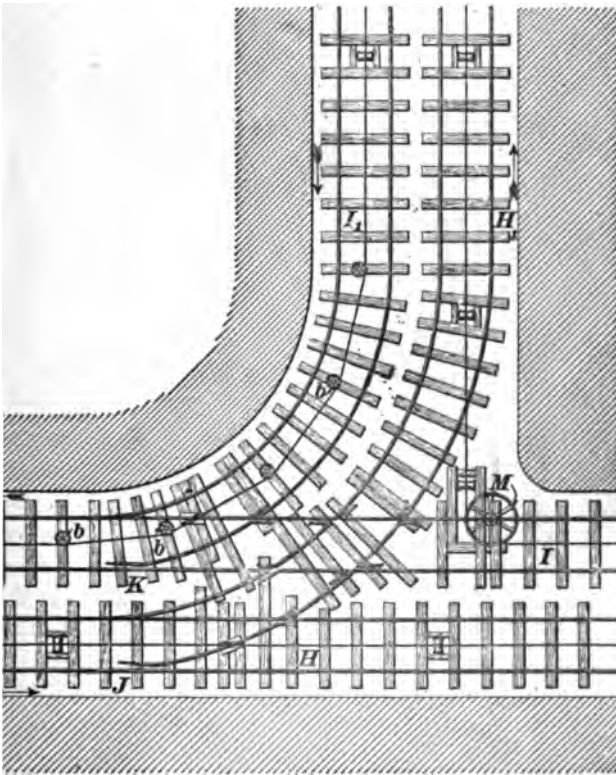
FIG. 861.

ȝuently tried, and has never given complete satis-
the rope is made continuous, both for the main

roads and the branching roads. To show the meaning of this, Fig. 861 is introduced. Looking at an arrow at the bottom of the figure near the letter *J*, it will be seen that the rope advances to its utmost limit into the mine, and then returns. When it reaches the entrance to a district, a wheel is fixed under the rails at *M* for deflecting the rope into the district. It now advances to the return wheel for that district, and then comes back, as shown by the arrow on the left-hand side of the figure. It then returns by the main haulage-road, as shown by the arrow a little above the letter *I*. For running cars into a district shown in the figure, the empty cars are turned in by the switch at *J*, which is in charge of a switch-keeper. The empty cars run on until they pass over the roller near H_1, where the grips are fixed to the rope. Then they proceed on their journey into the district workings. In the same way, when cars are returning by the main haulage-road, the connection of the grips with the rope is unmade at *I* near the deflecting wheel *M*, and the cars run with their acquired momentum until they nearly reach the roller *I* at the left side of the figure; here the connection is remade, and the loaded cars advance on their journey to the hoisting-shaft.

2385. The chief disadvantage of this system is, that the rope undulates so much with the varying tension that the cars travel unsteadily, and, in consequence, the rope is soon damaged. The unsteadiness arises from two causes: 1. On a long lead of rope resting on rollers sixty feet apart, the vibrations or undulations become deep and rapid. 2. The amount of elasticity increases with the length of the rope. These two causes give to the cars the jerky movement that has just been referred to. The worst of all is, that the longer this continuous band of rope is made, the heavier the rope must be for the increased traction, and the jerks due to such a heavy rope become stronger and more pronounced in their character than those produced by a light rope. The system of using a continuous band for a main and district haulage is not a good one; this becomes more evident

(a)

(b)
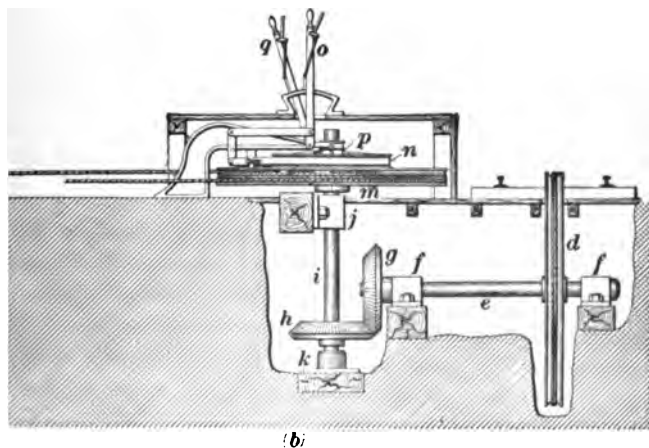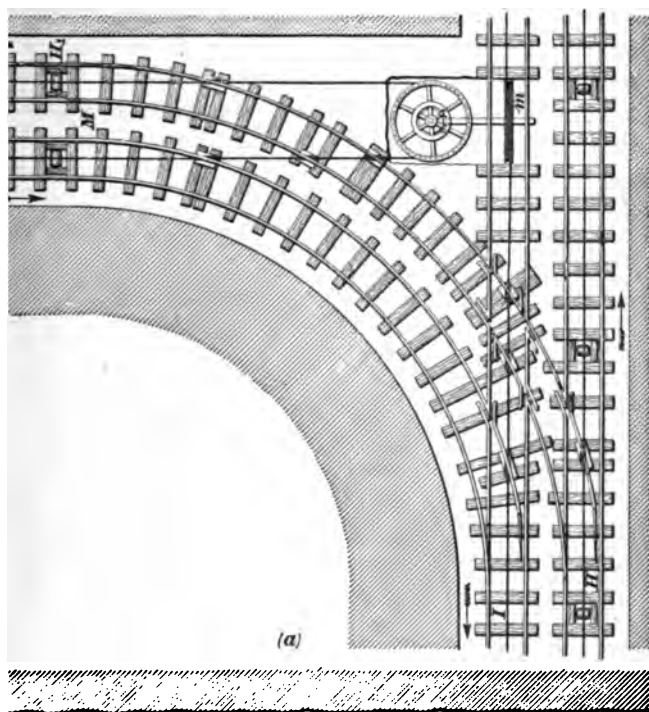
FIG. 864.

when compared with the system in which the haulage is done with a series of bands. When a district rope breaks, it only causes a local stoppage, whereas, when a continuous band breaks, the whole of the mine haulage is interrupted and stopped until the connection is remade.

2386. The endless-rope system that is worked with a series of bands is in practice preferable to all others. Its mode of action is shown in Fig. 862.

For the convenience of explanation, this mode of hauling is called the multiple-band system. Fig. 862 illustrates the character of the mechanism by which the secondary or district bands are actuated. In the figure it will be seen that at the junction of the district bands with the main ones, repeating fleet-wheels are used; for example, at *m*, Fig. 862 (*a*), the main haulage-band makes two complete turns round that fleet-wheel, and then continues its onward course. The wheel *d*, Fig. 862 (*b*), around which the main rope passes, drives the horizontal fleet-wheel *m* through the bevel-gears *g*, *h*.

The horizontal fleet-wheel is seen in the plan at the left-hand side of *m* in the plan of the roads. By this repeating fleet-wheel connection, a separate district band is made to haul the loaded cars out of *M* along the track I_1, and to haul in the empty cars along the track II_1. The levers *q* and *o* are applied to clutch the horizontal fleet-wheel in and out of gear. When no work is coming out of district *M*, the horizontal fleet-wheel is thrown out of gear, and when this district haulage is required it is put in gear. The advantage of this arrangement is that the stress on the rope of the main haulage-band is greatly reduced, for it seldom happens that all the district bands are moving at one time.

2387. Where endless-rope haulage is done by one engine, the stress on the main band of rope is very great, because this rope must sustain the stress due to the whole of the traction of the mine. Therefore, the rope of the main bands must be very large and heavy, or otherwise, after it has been in use a short time, it is liable to break and produce stoppages and delays of a very serious character. If light

opes are used on the main band, waste can be avoided by
aking them off after they have been used a short time, and
ısing them to do lighter work as district bands.   It is true
hat this can be done with heavy ropes, but the waste of
energy due to the use of heavy ropes may be at once per-
ceived when the student remembers that the weight of these
opes increases the traction due to friction, and thereby
argely augments the cost of generating the motive power.
t is, however, evident that if only one engine is employed
or a general endless-rope haulage in a large mine with a
arge output, all the difficulties here pointed out are unavoid-
ble.   They may, however, be prevented entirely, and the
ystem of haulage may be immensely simplified by using
lectric or compressed-air motors, or gasoline-engines, for
orking the district bands.   By this means, two great ad-
antages are secured.   First, when a district band is not run-
ng, there can be no waste of costly energy, and when the
pe is running the haulage can be done quickly or slowly,
cording to the exigencies of each special case, thus saving
needless waste of power.   Besides, the ropes can be of the
ghtest character, consistent with economy and efficiency,
d thus the tractive force can be reduced.   Perhaps the
llowing remarks will clearly illustrate the case in question :
Suppose there are 10 bands of rope all connected with re-
ating fleet-wheels; then the main bands are subjected to
e stress due to 10 sectional haulages.   Suppose, again, that
l the bands are worked by separate motors; then each rope
ill only require a strength sufficient to do its own special
ork, which will be one-tenth of the whole.   Consequently,
pes of one-tenth the strength will do the haulage as effi-
ently as the large rope previously noticed.   It is sometimes
ged that transmitted energy is attended with loss.   This
true, but it is far better to lose 25 or 30 per cent. of the
otive steam-power to obtain the advantage of transmitted
ergy by which separate motors are used for each band of
pes, because the tractive force is reduced by the reduction
. the weights of the ropes, and the saving of energy is in-
eased by reducing the velocities of the district bands to a

speed within the compass of the work to be done. By using gasoline-engines for the district haulages, however, there is no loss due to transmitted energy, because the energy required is generated in the cylinder of the engine.

2388. Extensive systems of endless-rope haulage are sometimes used on the surface. When, as sometimes is the case, the track is not less than two miles in length, and a great weight of coal must be carried over the road, it is necessary that the cars should be set at relatively short distances apart. The result is that a heavy tractive force falls on the rope. It can scarcely be urged that, in a case like this, a continuous lead should be broken for reducing the weight of the rope. In a mine, however, this reasoning does not apply, because there a single band can not be used, as separate bands are required for each of the districts into which the mine is divided, and as the division is imperative, all the advantages of a multiple haulage can be secured by using separate motors for each band. To do these separate haulages, however, electricity or compressed air must be used to transmit energy, or gasoline-engines must be used. Which of these three plans may be preferred depends upon the conditions met with in different mines. For example, the transmission of electric energy would sometimes take precedence. This being the case, it might be suggested that it would be wise to dispense with the ropes altogether, and introduce in their stead electric or compressed-air locomotives. They could not, however, be used in a large mine with a large output, because the trains running in and out of the different districts would interrupt one another's free passage, and cause great delays in the contracted roads of a mine, or in mines where the haulage is very long, and principally on one or two roads. The same may be said of compressed-air locomotives. Locomotives have their place in small mines, but for a large output, in a large mine with a number of districts, no system of haulage has yet been introduced that furnishes equal facilities with that of the endless rope. It does its work cheaply and efficiently within the limits of the ordinary working

hours. Locomotive haulage in large coal-mines worked in numerous districts is altogether out of the question, but electric or compressed-air motors or gasoline-engines for the main and district haulage give to the endless-rope system merits that are unobtainable by any other means.

2389. Before considering the mathematical questions that arise in reference to endless-rope haulage, there are certain details in the appliances of the system that are worthy of attention, and should be somewhat comprehensively understood by the mining student. These will now be described.

TAIL OR RETURN SHEAVES.

2390. As previously stated, tail or return sheaves are fixed at the inner ends of the tail-rope haulage-roads for reversing the direction of the rope and leading it back to the hauling-drum. Similar sheaves for the same purpose are used for the various bands of an endless-rope haulage system, but in this case the sheaves are mounted on cars, and are known as tension or balance cars. Their mode of action will, however, be taken up under a separate head.

2391. Fig. 863 shows a horizontal tail-sheave which may be used for the engine-plane or tail-rope systems. The cast-iron spider m is firmly bolted to the timbers shown, and is provided with a steel or wrought-iron pin n, around which the sheave o revolves. That part of the pin which passes through the spider has a smaller diameter than that which passes through the sheave, so that, by screwing up the nut, it is firmly held in the spider without clamping the hub of the sheave, and, therefore, offers no resistance to the free turning of the sheave. To prevent the sheave from coming off, and to keep dirt from entering the bearing, the pin is provided with a nut and washer p. For properly lubricating the sheave, the pin is provided with an oil-cup q, the oil from which flows through the hole of the pin, as shown by the dotted lines.

The sheave should be rigidly anchored, so that there is no

possibility of it becoming loose under a heavy load. The
best method of doing this is to bolt the spider to a masonry
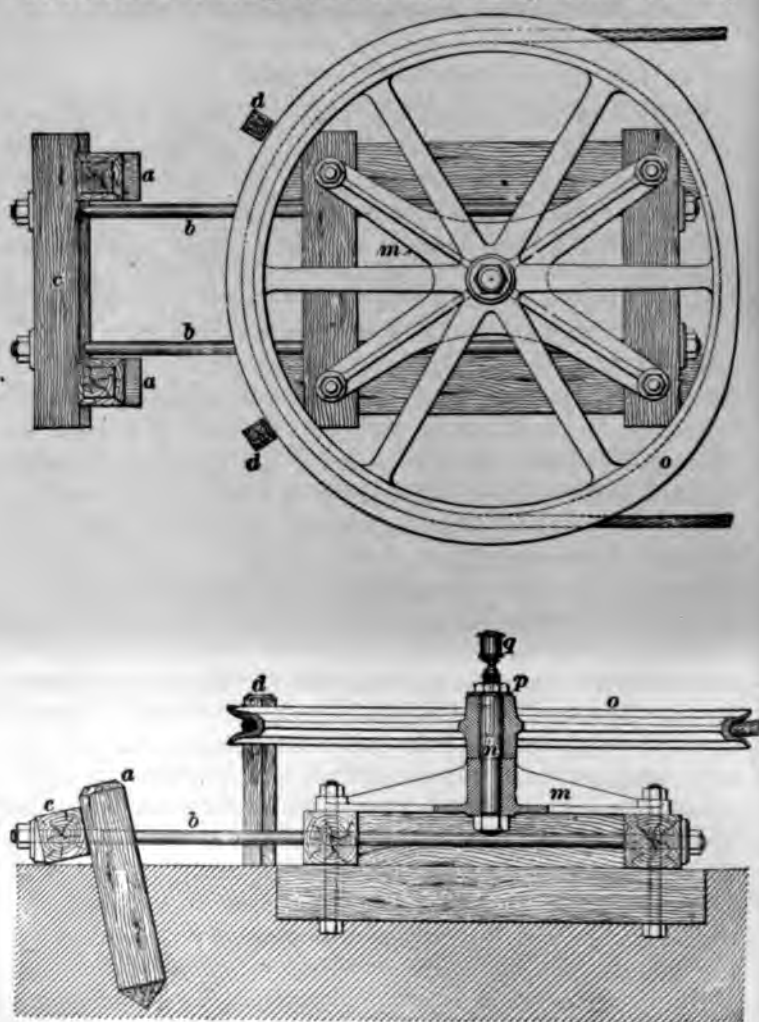
FIG. 863.

foundation built especially for it. This, however, is not
always possible, owing to varying conditions; and even if it

were, it would be too expensive, unless the system were to be a permanent one.

In the figure is shown a common method of anchoring a sheave. Here the foundation is built of 12″ × 12″ timbers, to which the spider is bolted, the bolts passing through both timbers as shown. At a distance of about three or four feet from the sheave is firmly secured in the ground, at an angle, two timbers *a* and *a*, to which the sheave is tied by the rods *b* and *b*. One end of each of these rods passes through the upper foundation timbers, and the other through a 12″ × 12″ timber *c* placed on the outside of the two timbers *a* and *a*. The ends of the rods are supplied with nuts and cast-iron washers. For preventing the rope from falling off the sheave when it becomes slack or when the stress is removed, two timbers *d* and *d* are firmly fastened in the ground close to the sheave.

2392. Tail-sheaves are sometimes arranged as shown in Fig. 864. Here, the sheave *O* is in a vertical position
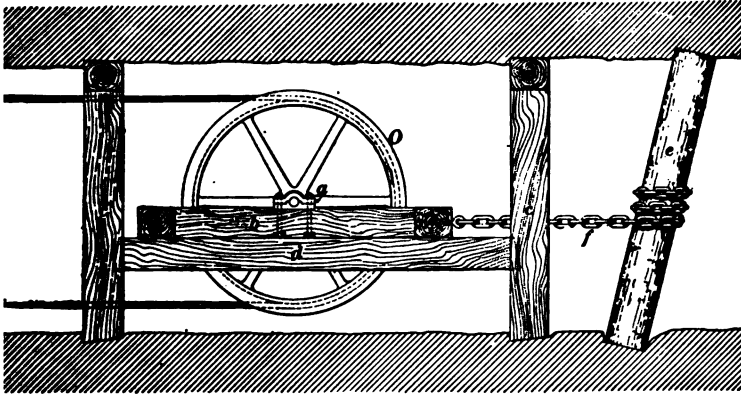
FIG. 864.

and revolves in bearings *a* placed on each side. It is firmly bolted to a strong rectangular frame *b* made of heavy timbers. The sheave is elevated a certain height by a wooden structure consisting of timbers *c* and *c*, rigidly fastened between the bottom and roof of the entry, and the horizontal timber *d*, as shown. This sheave is anchored by

rigidly fastening a post *e* at an angle, and wrapping one end of a chain *f* several times around it, the other end of which is fastened by means of an eye-bolt to the frame-timber *b*.

2393. In Fig. 865 is shown another method of anchoring a vertical tail-sheave. Here, two heavy timbers *a*, which
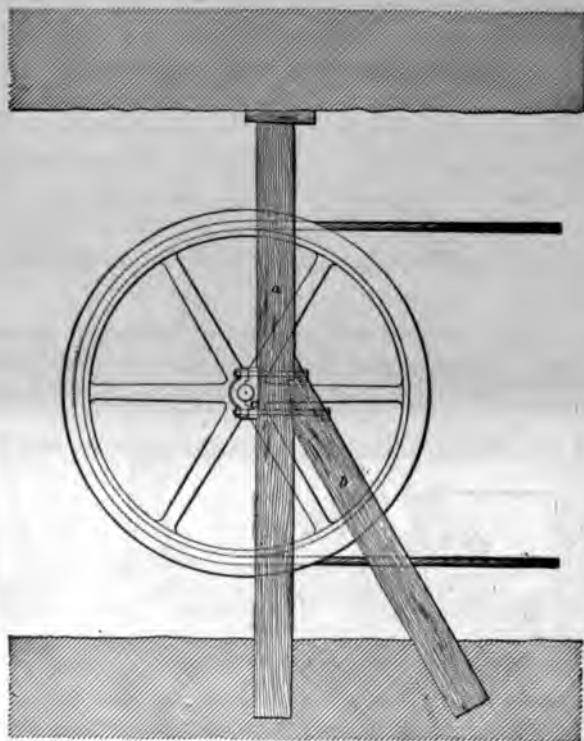
FIG. 865.

are strengthened by braces *b*, are rigidly fastened in a vertical position on each side of the sheave. The joint formed by the timbers and braces should be so made that there will be no liability of the brace slipping when under a great stress. This may be done as shown in the figure. The bolts which hold in place the bearings *c*, in which the sheave revolves, pass through the vertical timbers and the braces *b*.

³y making the joint as shown, the bolts are wholly re-
ieved from vertical stress due to the brace *b*.

2394. When it becomes necessary to place the tail-
heave directly under the track, the arrangement shown in
'ig. 866 is generally used. A pit *L* of sufficient depth is
rovided, in which the tail-sheave *O* is placed. At a distance
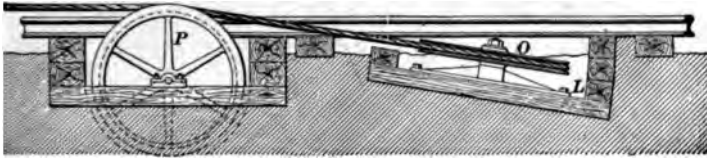
FIG. 866.

f about 30 or 35 feet from the tail-sheave is located a de-
ecting or guide sheave *P*. The rope, in coming along the
:nter of the track, is deflected by this sheave, and then runs
·ound the tail-sheave as shown.

2395. Tail and deflecting sheaves, around which a rope
: 19 wires to the strand is to be run, should have a diameter
: not less than 60 times the diameter of the rope; if the rope
is 7 wires to the strand, its diameter should be not less
ian 100 times the diameter of the rope.

2396. When locating guiding or deflecting sheaves
·ound which a rope is to be led, it is best to locate them in
ıch a manner that the angle formed by the two parts of the
ıpe which pass around the sheave is as large as possible.
nce the greater the angle the smaller the stress on the
:arings in which the sheave revolves. That this state-
ıent may be clearly understood, reference should be made
› Fig. 867, which represents an engine-plane. Here the
ıgine is located at *A* and the cars are at *B*. In order to pull
ıe cars in the direction indicated by the arrow (and for
:her reasons), it was necessary to locate three sheaves, as
ıown in the figure.
Assume that the resistance offered by the cars is 5 tons;
ıen, neglecting the friction of the sheaves and the rope,

the tension in all parts of the rope will be 5 tons. To deter-
mine the load on the pin of the sheave C, around which it
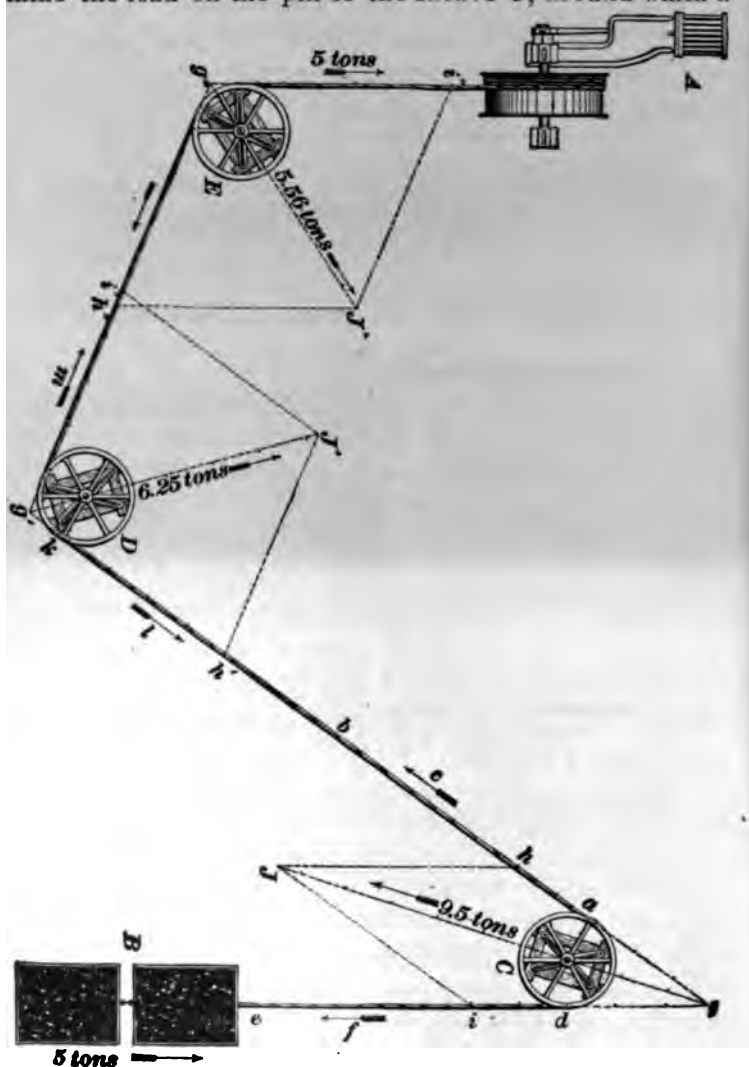
FIG. 867.

revolves, proceed as follows: To move the cars B, a force of
5 tons must be applied to the part $a\,b$ of the rope in the

direction of the arrow *c*.   There is also a force of 5 tons act-
ing in the portion of the rope in the direction of the arrow *f*
due to the resistance of the cars.   The sheave *C*, owing to
these two forces in the rope, tends to move towards *J* on the
line *g J*, but, as a matter of course, is prevented from doing
so by the pin around which it revolves, and by the anchor-
ing.   The force which tends to move the sheave *C* towards *J*
along the line *g J* may be found by producing the center
lines of the portions *a b* and *d e* of the ropes until they inter-
sect at *g*.   To any convenient scale, lay off from the point
of intersection *g* the points *h* and *i* on the portions of the
rope *a b* and *d e*, respectively, each equal to 5 tons.   Through
the point *h* draw the line *h J* parallel to the portion of the
rope *d e*, and through the point *i* draw the line *i J* parallel
to the portion of the rope *a b*; then, draw the diagonal *g J*
of the parallelogram.   This diagonal is the resultant, and
represents the greatest force acting on the sheave *C*.
Measuring it to the same scale that has been used to lay off
the forces *g h* and *g i*, respectively, its value is found to be
9¼ tons.   This 9¼ tons represents the pressure on the journal
and on the bearing of the sheave, and is nearly twice the
stress in the rope.   The pressure acting on the sheaves *D*
and *E* is found in a similar manner.

In ascertaining the pressure on the sheave *D*, the direction
of the stress in the portion of the rope *k b* must be consid-
ered as acting from *k* towards *b a*, as shown by the arrow *l*;
that is, it must be regarded as being produced by the resist-
ance of the cars.   The direction of the stress in the portion
of the rope *g' g''* due to the pull of the engine is from *g'*
towards *g''*, or in the direction of the arrow *m*.   Construct-
ing the parallelogram of forces as above, and remembering
that the tension in the rope still remains five tons, the re-
sultant is *g' J'*, and measuring it to the same scale as has
been used to lay off the forces *g' i''* and *g' h'*, respectively,
it is found to scale 6¼ tons, a result considerably less than in
the case of the sheave *C*.   It will also be noticed that the
angle *i'' g' h'*, formed by the rope led around the sheave *D*, is
considerably greater than the angle *h g i* formed by the rope

at the sheave C. In a similar manner is found the resultant stress $g'' J''$ on the sheave E; measuring it to the scale used to lay off the force $g'' h''$ and $g'' i''$, respectively, it is found to equal about 5.56 tons. In a similar manner the resultant stress on any sheave may be found. It matters not whether the sheave be placed in a vertical, horizontal, or angular position.

The above results indicate the best method of placing the spiders to resist the stress on the sheave; namely, as shown in the figure, with the long side of the timbers which support the spider parallel to the resultant $g J$, $g' J'$, or $g'' J''$.

From the above, it is quite evident that a sheave may be used very satisfactorily in one place and not at all in another. Thus, referring to Fig. 867, a sheave placed at E would be subjected to a load of only 5.56 tons, while, if it were placed at C, it would be subjected to $9\frac{1}{2}$ tons, or would have about double the load on it. The sheave placed at E could be considerably lighter than the one placed at C.

TENSION, OR BALANCE, CARS.

2397. A new wire rope permanently elongates or stretches, on account of the twist of the strands, from about 1 to $1\frac{1}{2}$ per cent. of its total length. Thus, a rope 1 mile, or 5,280 feet, long of an endless-rope haulage plant will continually become longer until it obtains a length of about $5,280 + 5,280 \times .015 = 5,359.2$ ft., or it increases in length about 79.2 ft. There will be a continual variation in the length of the rope due to the temperature of the atmosphere, which will cause the rope to expand or contract. A steel-wire rope will expand or contract .00000599 of its length, and an iron-wire rope .00000686 of its length, for each degree of change in temperature; hence, if the above rope were of steel, it would be $5,359.2 \times (100 - 32) \times .00000599 = 2.183$ ft. longer in summer when the thermometer stands at 100° than in winter at 32°; and an iron-wire rope would be $5,359.2 \times (100 - 32) \times .00000686 = 2.5$ ft. longer. Were no provisions made for this increase and decrease in the length of the rope, the rope would be slack at one time
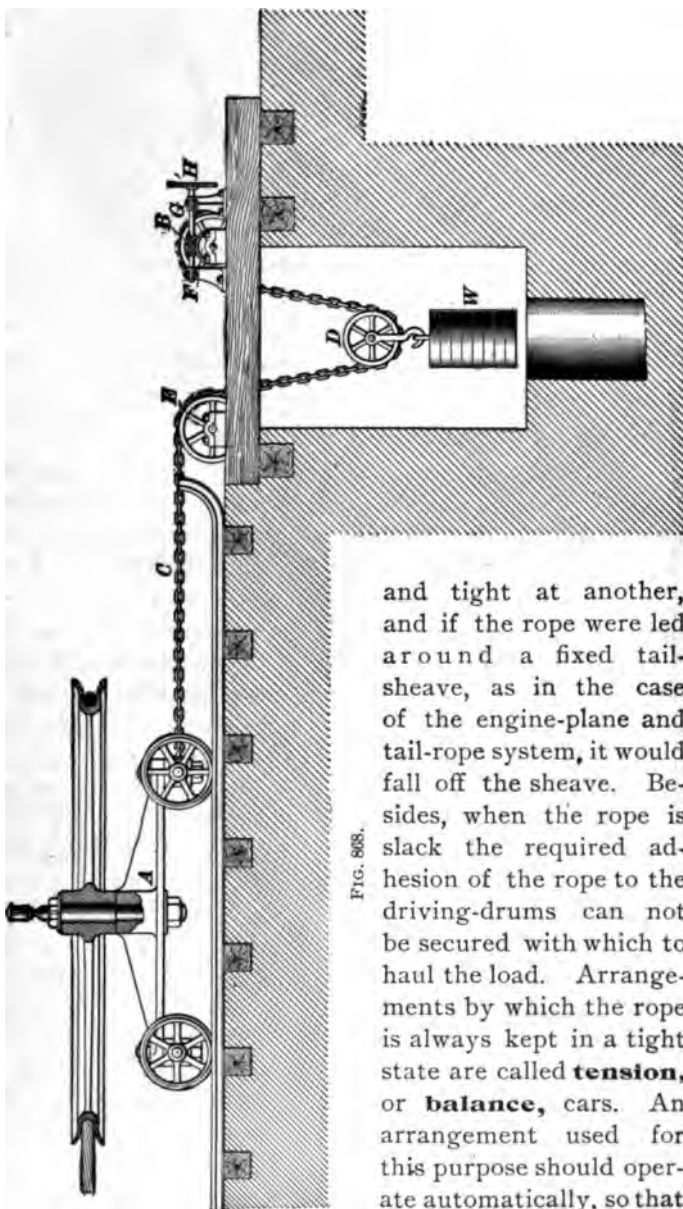
Fig. 868.

and tight at another, and if the rope were led around a fixed tail-sheave, as in the case of the engine-plane and tail-rope system, it would fall off the sheave. Besides, when the rope is slack the required adhesion of the rope to the driving-drums can not be secured with which to haul the load. Arrangements by which the rope is always kept in a tight state are called **tension,** or **balance,** cars. An arrangement used for this purpose should operate automatically, so that
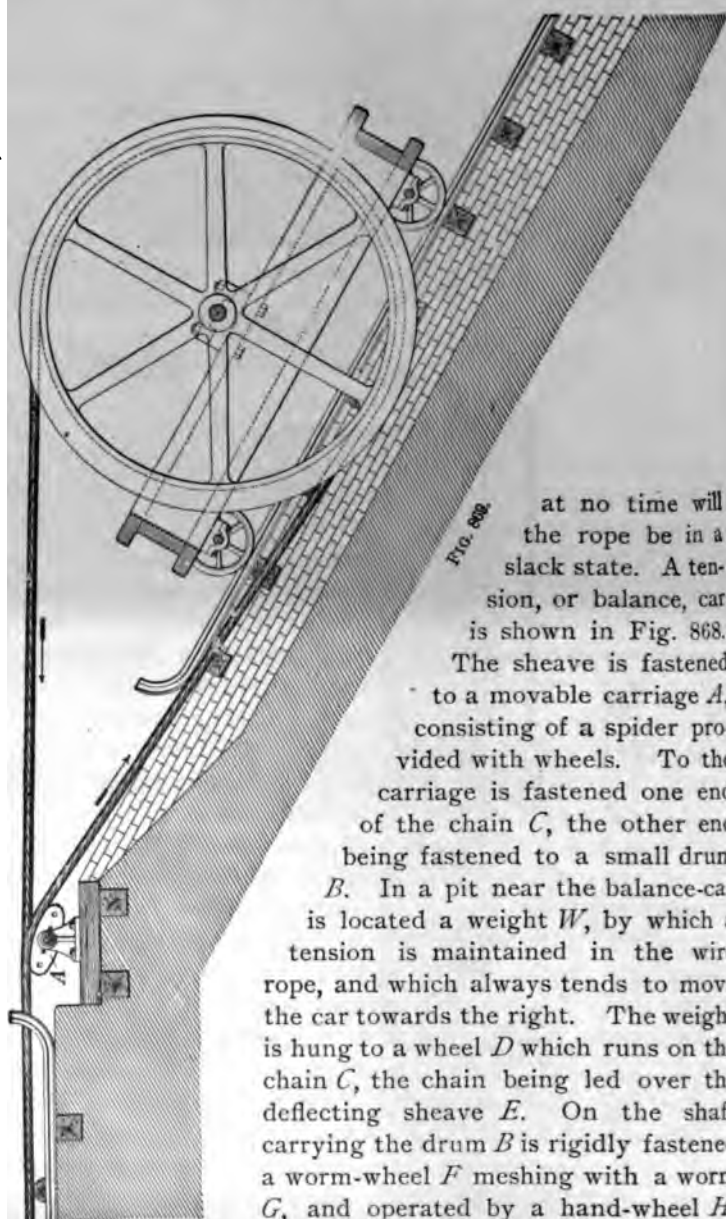
FIG. 868.

at no time will
the rope be in a
slack state.   A ten-
sion, or balance, car
is shown in Fig. 868.
The sheave is fastened
to a movable carriage *A*,
consisting of a spider pro-
vided with wheels.   To the
carriage is fastened one end
of the chain *C*, the other end
being fastened to a small drum
B. In a pit near the balance-car
is located a weight *W*, by which a
tension is maintained in the wire
rope, and which always tends to move
the car towards the right.   The weight
is hung to a wheel *D* which runs on the
chain *C*, the chain being led over the
deflecting sheave *E*.   On the shaft
carrying the drum *B* is rigidly fastened
a worm-wheel *F* meshing with a worm
G, and operated by a hand-wheel *H*.

After the wire rope has expanded enough to cause the weight
W to hang quite low in the pit, the weight is raised by
turning the hand-wheel H, which causes the drum to revolve
through the intervention of the worm and the worm-wheel,
thus shortening the chain C. The weight W in this case
has a movement of only one-half that of the carriage, but
must be twice as heavy as one that simply hangs at the end
of the chain. The rails on which the balance-car travels
should be turned up at their ends, as shown in the figure,
so as to form an obstruction beyond which the car can not
pass. It is well to have a second rope or chain lying loose,
having one end fastened to the car and the other to the
drum B, so that in case the first breaks the second takes its
place.

2398. In Fig. 869 is shown another arrangement of a
tension-car. A heavy car carrying the sheave is placed on
a short inclined plane. The rope coming along the center of
the track is carried over the deflecting sheave A, then passed
round the sheave of the balance-car, and is then led back
over the road. The inclination of the plane and the weight
of the balance-car are such that the proper tension in the
rope is maintained by the tendency of the car to descend
the plane; any expansion or contraction of the rope is
immediately taken up by the car.

2399. When it is required to provide tension arrange-
ments for the expansion and contraction of the rope only,
the permanent stretch of the rope being taken up by other
means, the arrangements shown in Figs. 870 and 871 may be
used. In Fig. 870, the rope is led over two sheaves A and
B, firmly held by suitable supports. In the center, between
these sheaves, is erected timber-work, so constructed that a
third sheave C can work up and down. The sheave C has
a weight W hung to it, and therefore causes a tension in the
rope. When the rope expands, the sheave C and weight W
descend, and when it contracts they ascend.

In Fig. 871 a somewhat different arrangement is shown.
Here, cast-iron standards having suitable bearings are

provided for the sheaves A and B.   At the upper extremity of
the larger standard is located a lever C, carrying a sheave D
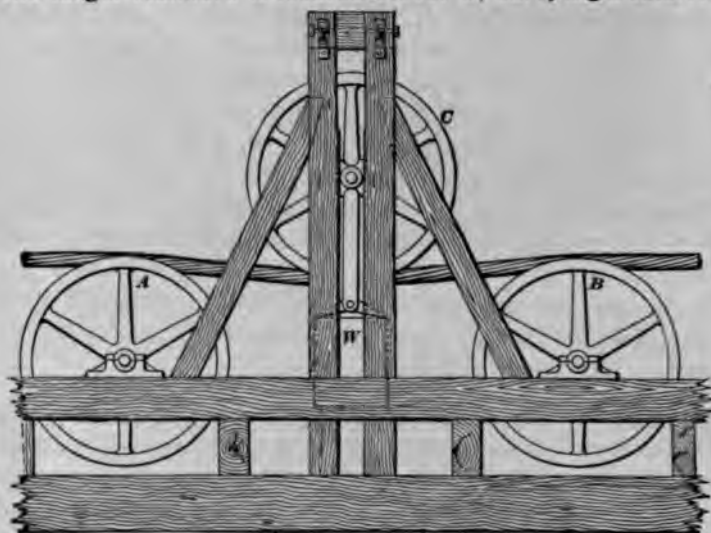
FIG. 870.
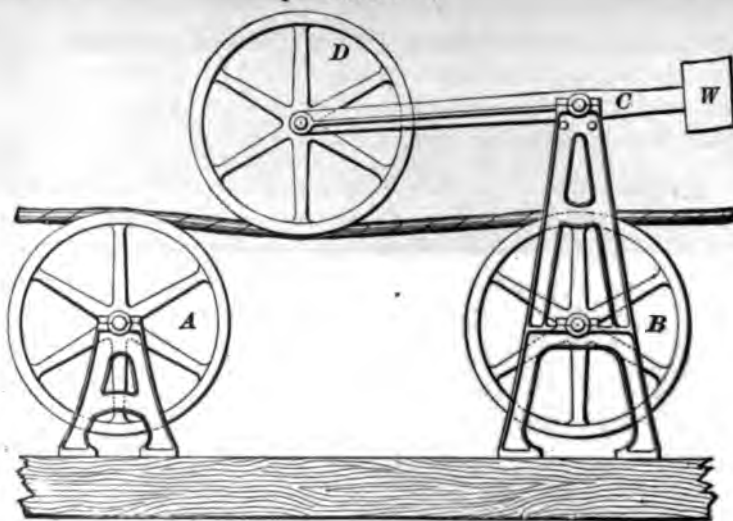
FIG. 871.

at one end and a counterpoise W at the other.   The counter-

poise is so constructed that it can be moved along the short arm of the lever, so that the tension in the rope can be properly adjusted.

2400. The rope of an endless-rope haulage system should always be led around a tension or balance car located at the far end of the road. Sometimes the rope is led around a tail-sheave firmly fixed, no means of a uniform tension in the rope being provided for. This is altogether wrong, and should not be practised.

A tension-car should always be placed close to the driving machinery. This is very essential for the proper working of the system, as otherwise it would be impossible to secure the requisite adhesion of the rope to the drum in order to haul the load. The slack or running-off portion of the rope should always be led around the sheave of the balance-car, for the slack can not be taken up on the loaded side.

CABLE-GRIPS FOR ENDLESS-ROPE HAULAGE.

2401. Fig. 872 furnishes an illustration of the mode of attaching a car to a cable in an endless-rope haulage system.
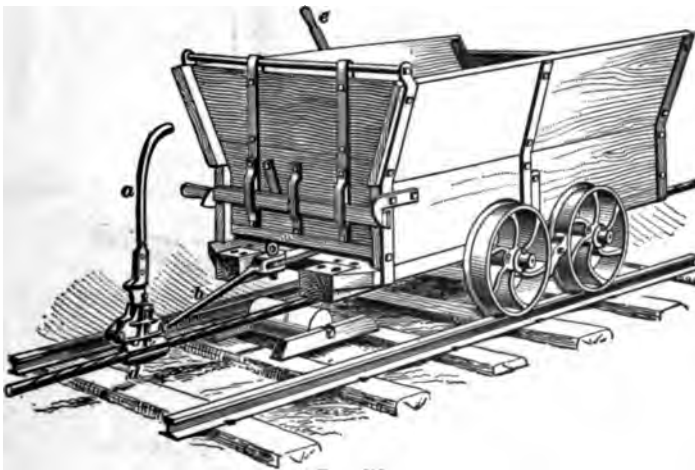
FIG. 872.

The grip, it will be seen, is coupled by a bar *b* to the car. At the moment the coupling is made, the lever *a* is lying

down on the coupling-bar *b*, so that, when *a* is raised, the jaws of the grips close, and immediately the car advances with the cable.   To disconnect the car, the lever *a* is pushed down, the jaws are released, and the car is brought to rest. Sometimes, however, its momentum must be destroyed by inserting sprags in the wheels.

2402.   One of the simplest of the grips is the old-fashioned grip-tongs shown in Fig. 873.   It consists of two jaws *a* and *b*, having handles *a'* and *b'* respectively.   The jaws

FIG. 873.

are held together by a pin *c*, having a nut on its other end. That part of the pin which passes through the handles is of smaller diameter than the part to which the chain *d* is fastened.   A shoulder, therefore, is formed on the pin which prevents any side movement of the jaws.   The grip is fast-

ıed to the car by the chain *d*, the hook being attached to
ıe draw-bar of the car. The jaws of the grip have a
ʼescent shape, so that the cable can be held between
ıem. The friction between the jaws of the grip and the
ıble being greater than the resistance of the car, the latter
 moved along with the speed of the cable. To obviate the
ıcessity of continuously holding the jaws together so as to
ıul the load, a link *e* is wedged over the handles. Not-
ithstanding that this grip is very simple in construction,
 is found to be very cumbersome; it can only be used for
ıuling light loads.

2403. Where heavy loads are to be hauled, it is neces-
ıry that the construction of the grip be such that a greater
ʼessure can be exerted on the cable by the jaws of the grip
ıan in that shown in Fig. 873. A grip which will do this,
ıd one which is extensively used, is shown in Fig. 874.
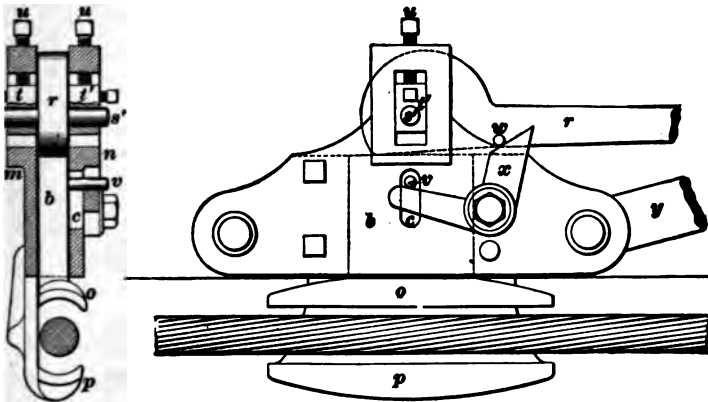
FIG. 874.

his consists of two plates or frames *m* and *n* bolted together.
ach frame has on its inner side a channel; these lie oppo-
te to each other, forming a recess in which the shank *b* of
ıe upper jaw *o* can slide up and down. The lower jaw *p*
 part of the frame *m*, and, therefore, remains stationary
 relation to the frames *m* and *n*. Between the upper
:tremities of the frames is provided a lever *r* for operating
ıe sliding-jaw *o*. This lever has trunnions *s* and *s'* which

24—31

work in bearings *t* and *t'* fitted to the frames.   That end or
part of the lever which works between the frames is of such
a shape that, when the lever is swung over to the opposite
sides, it forces the jaw *o* downwards, and causes it to grip the
cable firmly.   After the jaws *o* and *p* are worn, so that the
pressure required to haul the load can not be secured by
swinging the lever *r* to the opposite side, the end of the
lever between the frame is lowered by forcing the bearings
t and *t'* downwards by means of the set screws *u, u,* after
which the cable can again be gripped firmly.   To cause the
jaw *o* to clear the cable after the grip has been released, a
pin *v* is fastened to the shank *b* of the jaw *o*, which projects
through an elongated hole *c* in the frame.   Firmly secured
to the lever *r* is another pin *w*, and to the frame *n* is pivoted
a bell-crank *x*, having the upper edge of the arm, which is
nearly perpendicular, inclined.   As the lever *r* is raised to a
vertical position, the jaw *o* is moved downwards, also the pin
v, thereby causing the bell-crank *x* to swing on its center.
In returning the lever *r* to its original position, the pin *w*
strikes the nearly perpendicular arm of the bell-crank *x*,
causing it to take the position shown in the figure, thereby
raising the jaw *o* by the pin *v*.   The grip is made of steel,
excepting the jaws *o* and *p*, which are lined with soft metal;
these, when worn out, can easily be replaced.   This grip is
fastened to the car by a rod having a hook at one end, part
of this bar being shown at *y*.

   2404.   When the cars are run singly, each car must be
provided with a grip; but if they are run in trains, a power-
ful grip may be attached to the first car, or the grip may be
mounted on a special car, to which the whole train of cars
may be attached.   The special car carrying the grip is called
the grip-car, and the grip is applied to the rope by either a
combination of levers or a hand-wheel.   In Fig. 875 is
shown a grip-car, which consists of a strong timber-frame
mounted on wheels.   The grip is fixed in the center of the
car in such a manner that no longitudinal movement can
take place; it may, however, be moved sideways automatic-
ally, in order to clear the guiding-sheaves when passing

ınd curves. The car is provided with brakes *b* and *b*
stopping the car more quickly, and to prevent any move-
ıt of the train on slight inclines when the grip is released
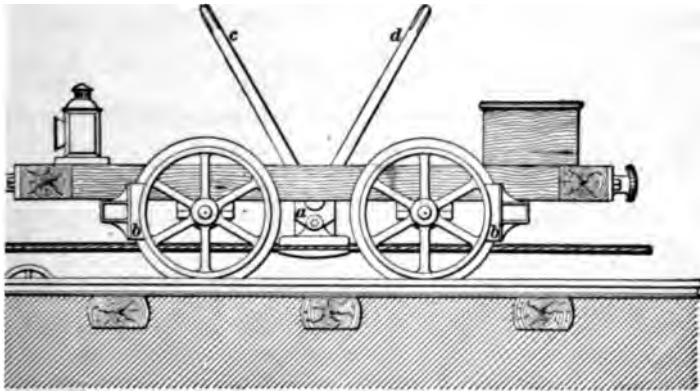
FIG. 875.

n the rope. The grip is operated by the lever *c*, and the
ɛes by the lever *d*; it is placed sufficiently high to clear
track-rollers.

.CULATIONS RELATING TO ENDLESS-ROPE HAULAGE.

**405. The Number of Cars and the Distance
ɛy Are Set Apart on a Main Haulage-Band.**

ɛt O = the output of coal in tons per day;

o = the weight of coal in tons a single car will carry;

o_1 = the weight of coal in the cars attached to the
haulage-band;

D = the distance traveled by a point in the rope in
one day;

d = the distance traveled by the rope from the return
sheave to the hoisting-shaft;

n = the number of full cars on the main-rope band;

r = the distance between cars on the rope.

Then,
$$o_1 = \frac{O\,d}{D}. \qquad \textbf{(206.)}$$

$$n = \frac{O\,d}{D\,o}. \qquad \textbf{(207.)}$$

$$r = \frac{d}{n}. \qquad \textbf{(208.)}$$

To understand these expressions, notice that if d is the distance a point in the rope moves through in passing from the return wheel to the hoisting-shaft, and that if D is the distance a point would move through in a day, supposing it always to be traveling shaftwards; then, $\frac{d}{D}$ will represent the fraction of the coal hauled in one outward journey of the point; that is, $o_1 = \frac{O d}{D}$.

Again, $n = \frac{o_1}{o}$, because if the weight of coal hauled in one journey is equal to o_1, and o is the weight of coal one car will carry, then $\frac{o_1}{o}$ is equal to the number of loaded cars on the rope, for the rope will throw off at the shaft that number for each distance.

Hence, $n = \frac{O d}{D o}$ and $r = \frac{d}{n}$, the distance the cars are apart on the rope, for if d is the distance from the return-sheave to the bottom of the shaft, and n is the number of cars on the main rope, $\frac{d}{n}$ must be equal to the distance the cars are apart.

EXAMPLE.—It is intended that the output of a mine shall be 3,000 tons of coal per day. The length of the road from the return wheel to the shaft is 6,600 feet, the amount of coal a car carries is 1 ton, the velocity of the rope is 3 miles an hour, and the time of one day is 10 hours. If the mean time of running, when allowance is made for stoppages, is 90 per cent. of the given working day, what will be the number of loaded cars on the main haulage-band, and what will be their distances apart, the weights being given in long tons?

SOLUTION.— $O = 2,240 \times 3,000 = 6,720,000$ lb.; $o = 2,240$ lb.; $d = 6,600$ feet; $D = 5,280 \times 3 \times 10 \times .90 = 142,560$ ft. Hence, by substituting these values in formula **207**, we have

$$n = \frac{O d}{D o} = \frac{6,720,000 \times 6,600}{142,560 \times 2,240} = 188.888, \text{ or } 189,$$

the average number of cars on the main-rope band. **Ans.**

From formula **208**,

$$r = \frac{d}{n} = \frac{6,600}{188.888} = 47.52 \text{ ft.,}$$

the mean distance the loaded cars are set apart on the rope. **Ans.**

2406. To realize the meaning and value of the equations in the last example, consider another example similar to the last, except that the length of the main band, or what is the equivalent of that, the length of the main haulage-road, is changed. Under such conditions it will be found that, although the length of the band is shortened and the number of cars on it is less, the distance they are apart is the same.

EXAMPLE.—Suppose that in the example given in Art. **2405** the length of the road from the return wheel to the shaft was 3,300 feet, instead of 6,600 feet, what would be the number of loaded cars on the main haulage-band, and how far apart would they be placed?

SOLUTION.—Applying formula **207**, we have

$$n = \frac{O\,d}{D\,o} = \frac{6,720,000 \times 3,300}{142,560 \times 2,240} = 69.444 =$$

average number of cars on the main-rope band. **Ans.**

Also, by applying formula **208**, we have

$$r = \frac{d}{n} = \frac{3,300}{69.444} = 47.52 \text{ ft.} =$$

the distance the loaded cars are apart on the rope. **Ans.**

From the above example, it is plain that, by halving the length d, the number of cars on the rope is halved, and yet the distance the cars are apart remains the same. From these facts an important point is learned; that is, that if none of the factors are altered but the length, a long band delivers as many cars at the hoisting-shaft as a short one. That is, if the cars are set at the same distances apart on the rope, and the velocity remains the same, the output is alike for all distances. This principle of action is the same, however, in relation to gravity-planes, engine-planes, and main and tail rope haulages; for if the lengths of the trains are made proportional to the lengths of the haulage-roads, then with the same velocity equal weights are hauled to the shaft in equal times. Consequently, when the length of the endless-rope haulage is doubled, and the number of cars on the band is doubled, and where the length of a main and tail rope haulage is doubled, and the number of cars in a train are doubled, if the velocity remains the same, the output will be the same in equal times in each case.

2407. The Weights of Haulage-Ropes in Relation to Their Velocities.—*For equal outputs in equal times, the weights of the haulage-ropes* vary inversely as the velocities. For example, when the velocity is doubled, the weight of the rope is halved, for two reasons: (1) the safe working loads of ropes vary directly as their weights; (2) as the velocity of a rope increases, the load on it is reduced; that is, when equal amounts of work are done in equal times.

Let l = the load or tension in the rope;

v = the velocity;

u = the units of work done.

Then $U = l\,v.$ **(209.)**

EXAMPLE.—In one case a tension of 4,000 pounds on a haulage-rope is required to move a train of 25 loaded cars with a velocity of 20 miles an hour, and in another case a tension of 8,000 pounds is required to move 50 loaded cars with a velocity of 10 miles an hour. Prove that if the diameter of the rope that hauls with a velocity of 20 miles an hour is 1 inch, the diameter of the rope that hauls with a velocity of 10 miles an hour must not be less than 1.414 inches.

SOLUTION.— $4,000 \times 20 = 8,000 \times 10$; therefore, U in one case is equal to that of the other.

Again, as the tension on a rope is equal to the load, and as the safe working load of a rope varies as its transverse or sectional area or its weight per unit of length, it follows that the diameters of ropes will vary as the square roots of the loads or sectional areas. Then, if the diameter of the rope for a load of 4,000 pounds is 1 inch, the diameter of the rope for a load of 8,000 pounds will be

$$\sqrt{\frac{8,000}{4,000}} = 1.414 \text{ inches. Ans.}$$

This means, also, that if equal weights of coal are hauled in different numbers of trips through equal distances in equal times, the diameters of the ropes vary inversely as the square roots of the numbers of trips. To make this clear, suppose one engine A hauls out four trips while another B hauls out one, and that in 10 hours they both haul out 1,000 tons through a distance of 5,000 feet; then the tension on the A rope is only one-fourth of that on the B rope, because A's load is only $\frac{1}{4}$ that of B's. Therefore, the diameter of A's rope requires to be only $\frac{1}{2}$ that of B's, because $\sqrt{\frac{1}{4}} = .5$.

2408. From the conclusions here arrived at, it might be considered that fast running would secure considerable economy in hauling, for not only would the traction due to the friction of a heavy rope be reduced by hauling with one of much less weight, but first cost of ropes would be much reduced. But such is not the case, because another factor of stress is introduced by high speeds that is hardly felt in low ones. For example, when an endless-rope haulage is done with a velocity of even four miles an hour, the ropes soon become kinked, flattened, and permanently injured by the grips. To start a heavy car from a state of rest, and give it a velocity of 4 miles an hour, greatly strains the rope; therefore, for this velocity, ropes must be made larger than those required for the stress due to ordinary traction. Some idea of this stress may be obtained by noticing that it varies as the square of the velocity. The stress due to starting a car in motion at 4 miles an hour is equal to the square of 4, or 16, as contrasted with the stress due to starting one with a velocity of 3 miles an hour, which is equal to $3^2 = 9$. To further show the importance of this stress, let an endless-rope haulage be run at 6 miles an hour; for this speed, the number of cars on the rope in a given distance would be half the number required to be put on a rope moving with a velocity of 3 miles an hour. Then, from this point of view, the diameter of the rope subject to half the former load would for a velocity of 6 miles an hour be reduced to $\sqrt{\frac{1}{2}} = .707$ of its original diameter.

On the other hand, the stress due to setting the cars suddenly in motion on a 6-mile velocity, as contrasted with a 3-mile velocity, would be as the squares of the velocities; consequently, if the diameter of a rope for a 3-mile velocity were one inch, that for a 6-mile velocity would be

$$\sqrt{\frac{3}{6} \times \frac{6^2}{3^2}} = 2.828 \text{ inches,}$$ assuming that the tension on the rope due to traction is equivalent to the energy required to suddenly set a car in motion from a state of rest to that of a velocity of 3 miles an hour.

2409. From what has been shown, it is evident that the velocity at which the cars should run on an endless-rope haulage requires calculation. For example, if an endless rope has a velocity of 4 miles an hour, the damage done to the ropes by the grips is very serious; but there are other troubles that require notice. In the first case, it is not safe for a person who is inspecting the road to cross a double track of this character. Again, the cars leave the rope at too high a velocity, and are, therefore, not sufficiently under control at the period when they are detached. The best results of the endless-rope haulage are obtained with a velocity not exceeding 2 miles an hour; for then persons having · to work on the tracks, such as oiling the rollers, inspecting the roof, sides, and timber, and doing the necessary inspection and repairs of rails, ties, etc., are able to take care of themselves. This is an important matter, for great delay and expense is saved by proper inspection and repairs. It is true that a velocity of 2 miles an hour necessitates the use of heavier ropes for traction; but the damage to ropes produced by the car-grips is reduced to a minimum, and, indeed, becomes so small a matter that in practice its consideration may be neglected.

2410. These remarks refer to underground haulage, but on the surface a velocity of 3 miles an hour may be used with good results. This velocity should never be exceeded on a typical endless-rope haulage, even on the surface. Running trains of a number of cars together with endless rope is not worthy of much consideration; for if the student considers the heavy weights being suddenly jerked into motion by the rope, he will have some idea of the great destruction of ropes that must arise from an inert resistance of this character.

2411. In the English mines, the endless ropes generally lie on the tops of the cars, called tubs; in the United States, the ropes generally lie under the cars, and, in cases where this system of haulage is modified for hauling groups of cars in trains, the ropes are sometimes gripped to the sides of the cars.

2412. Tension on the Ropes and the Horse-powers of Endless-Rope Haulages.

Let $W =$ the weight of the loaded cars on one side of a rope band;

     $w_1 =$ the weight of the empty cars on the ingoing side of a rope band;

     $w =$ the weight of a rope band;

     $C = \frac{1}{40} =$ the coefficient of friction;

     $a =$ the mean grade per cent. ;

     $T =$ the tension in the rope in pounds.

The following expression is the equivalent of the tension on the haul-out side of the rope, and it is true for the same side of all the bands in the series:

$$T = \frac{(W + w_1 + w)}{40} + a\,(W - w_1). \qquad (210.)$$

Observe that the weights of the cars carrying the coal are balanced by the weights of the empty cars. Again, the weight of the rope on one side of the band is balanced by the rope on the other side of it, and, therefore, the only weight that develops a gravity force is that of the coal; hence, in equation **210,** $a\,(W - w_1)$ represents the gravity force.

EXAMPLE.—The track of a single endless-rope haulage-band is 1 mile in length, and the rope hauls out 800 long tons of coal in 10 hours. The cars carry 1 long ton of coal, and an empty car weighs 1,760 pounds. The velocity of the rope is 2 miles an hour, and the size of the rope is such that it weighs 2 pounds per foot of length. What is the tension on the rope and the horsepower of the hauling-engine, supposing the mean grade of the road to be level ?

SOLUTION.— O, the output, $= 800 \times 2,240 = 1,792,000$ lb.; o, the weight of coal carried by one car, $= 2,240$ lb.; d, the length of the track, $= 5,280$ ft.; D, the travel of the rope in 90 per cent. of 10 hours, $= 5,280 \times 2 \times 10 \times .90 = 95,040$.

Applying formula **207,**

$$n = \frac{O\,d}{D\,o} = \frac{1,792,000 \times 5,280}{95,040 \times 2,240} = 44.44, \text{ the number of loaded cars.}$$

$W = 44.44 \times 4,000 = 177,760$ lb.; $w_1 = 44.44 \times 1,760 = 78,214.4$ lb.; and $w = 5,280 \times 2 \times 2 = 21,120$ lb.

Applying formula **210,**

$$T = \frac{(W + w_1 + w)}{40} + a(W - w_1) = \frac{(177,760 + 78,214.4 + 21,120)}{40} + 0 =$$

6,927.86 lb., the tension required. Ans.

Observe that the expression $a(W - w_1)$ becomes 0 in this case, because the road is level, which makes $a = 0$.

Again, the horsepower can be found as follows:

v, the velocity in feet per minute $= \dfrac{5,280 \times 2}{60} = 176$ ft.

Substituting in formula **202,**

$$H = \frac{Tv}{33,000} = \frac{6,927.86 \times 176}{33,000} = 36.95 \text{ H. P. } \text{ Ans.}$$

EXAMPLE.—Let the given values be the same as in the previous example, except that the road has a mean up grade to the shaft of 2 per cent. The tension on the rope and the horsepower of the haulage engine is required.

SOLUTION.—Applying formula **210,**

$$T = \frac{(W + w_1 + w)}{40} + a(W - w_1) =$$

$$\frac{(177,760 + 78,214.4 + 21,120)}{40} + .02(177,760 - 78,214.4) = 8,918.27 \text{ lb.,}$$

the tension required. Ans.

Applying formula **202,**

$$H = \frac{Tv}{33,000} = \frac{8,918.27 \times 176}{33,000} = 47.56 \text{ H. P. } =$$

the horsepower required. Ans.

EXAMPLE.—Let all the values be the same as in the first example, except that the road is down grade to the shaft at the rate of 2 per cent. Find the tension on the rope and the horsepower of the engine.

SOLUTION.—Applying formula **210,**

$$T = \frac{(W + w_1 + w)}{40} - a(W - w_1) =$$

$$\frac{(177,760 + 78,214.4 + 21,120)}{40} - .02(177,760 - 78,214.4) = 4,936.45 \text{ lb.} =$$

the tension required. Ans.

Here the gravity force $a(W - w)$ is minus, because it acts towards the engine.

Applying formula **202,**

$$H = \frac{Tv}{33,000} = \frac{4,936.45 \times 176}{33,000} = 26.33 \text{ H. P., the required horsepower.}$$
Ans.

EXAMPLE.—All the values for a main haulage-rope band are the same as those given in the first example, and, like it, the track is level, and, therefore, the tension due to that rope alone is the same as first

found, namely, 6,927.36 pounds. Two districts bands A and B, however, deliver in the present case to the main band; the band A hauls out, along a track 769 feet long, ⅓ of the 800 tons. This road has an up grade to the main road of 3 per cent. The band B hauls out ⅔ of the 800 tons, along a road 2,312 feet in length, with a down grade to the main haulage-band of 4 per cent. What is the tension in the A and B ropes? What is the total tension on the main rope? What is the required horsepower to do the haulage, if the rope on the band A weighs .88 lb. per foot, and the rope on the band B weighs 1.2 lb.?

SOLUTION.—First find the number of loaded cars on the A band.

$O = 800 \times \frac{1}{3} \times 2,240 = 716,800$ lb.; $d = 769$ ft., and $D = 5,280 \times 2 \times 10 \times .90 = 95,040$ ft.

Applying formula **207**,

$$n = \frac{O\,d}{D\,o} = \frac{716,800 \times 769}{95,040 \times 2,240} =$$

$2.589 =$ the number of loaded cars on the band A.

To find the tension in this rope, formula **210** is applied.

$W = 4,000 \times 2.589 = 10,356$ lb.; $w_1 = 1,760 \times 2.589 = 4,556.64$ lb.; and $w = 769 \times 2 \times .88 = 1,353.44$ lb.

Then, $T = \dfrac{(W + w_1 + w)}{40} + a\,(W - w_1) =$

$\dfrac{(10,356 + 4,556.64 + 1,353.44)}{40} + .03\,(10,356 - 4,556.64) = 580.63$ lb., the tension in the A band. Ans.

Next, the number of loaded cars on the B band is found by again applying formula **207**. In this case, $O = 800 \times \frac{2}{3} \times 2,240 = 1,075,200$ lb.; $d = 2,312$ ft., and $D = 5,280 \times 2 \times 10 \times .90 = 95,040$ ft.

Then, $n = \dfrac{O\,d}{D\,o} = \dfrac{1,075,200 \times 2,312}{95,040 \times 2,240} = 11.6767 =$ the number of loaded cars on the band B.

The tension in this band is also found by formula **210**.

$W = 11.6767 \times 4,000 = 46,706.8$ lb.; $w_1 = 11.6767 \times 1,760 = 20,550.992$ lb., and $w = 2,312 \times 2 \times 1.2 = 5,548.8$ lb.

Then, $T = \dfrac{(W + w_1 + w)}{40} - a\,(W - w_1) =$

$\dfrac{(46,706.8 + 20,550.992 + 5,548.8)}{40} - .04\,(46,706.8 - 20,550.992) = 773.93$, the tension required for B. Ans.

Here the gravity factor is minus again, because it works with the engine. The total tension in the main rope is $6,927.36 + 580.65 + 773.93 = 8,281.94$ lb. Ans.

Finally, applying formula **202**,

$$H = \frac{T\,v}{33,000} = \frac{8,281.94 \times 176}{33,000} = 44.17,$$

the total horsepower required to run the engine. **Ans.**

GENERAL APPLIANCES.

WIRE ROPES.

2413. Having considered the different methods of haulage, we will now give a description of the various appliances more or less common to all of the above systems, and which have not already been considered in detail.

2414. The wire rope used in mine haulage is usually made of six strands of iron or steel wires laid around a central core of hemp. Each of these strands is made up of either 7 or 19 wires, and, therefore, a wire rope is made up of either 42 or 114 wires, the final size of the rope depending upon the size of the wire used. It is evident that a one-inch rope containing 42 wires must be made of larger wires than a one-inch rope containing 114 wires. The former rope, on account of the larger size of the individual wires, is well adapted to service involving much wear; while the latter rope, being made of smaller wires, is more pliable, and is thus adapted to service in which the rope is required to do much bending. Thus, in the tail-rope system, the main rope has 7 wires to the strand, and the tail-rope 19 wires to the strand. Ropes are made with either a short or a long twist, according to the use for which they are intended. Ropes with a long twist work smoothly over the rollers and sheaves, and stretch but little, thus adapting them for tail-ropes. The short twisted rope is much more elastic, and should be used where the rope is liable to sudden jerks, as in the case of main ropes on engine and tail-rope planes. The short twisted rope in such a case acts as a sort of a spring, and stands without injury a shock which would break the more rigid long twisted ropes. The lay of the wire is generally opposite to the lay of the strands, although a rope has been recently introduced with the lay of both strands and wires in the same direction.

2415. A style of rope known as locked-wire rope has been introduced by some manufacturers. This rope has a

ooth surface approaching that of a round bar, in conseence of which it wears longer than the ordinary ropes, and, sides, does not wear out so quickly the wheels over which is led; it also has a greater strength than the ordinary re ropes, but has one disadvantage, in that it can not be liced.

2416. The strength of the ordinary wire rope is about ⅘ of the strength of the individual wires composing it, the re losing about 25% by bending. The working strength a wire rope may be taken at from one-fifth to one-seventh e breaking strength.

2417. Tables 46 and 47 give the diameters, weights, and eaking loads of iron and steel wire ropes of six strands und around a hemp center, having 19 and 7 wires to the and.

TABLE 46.

ON AND STEEL WIRE ROPES, 19 WIRES TO THE STRAND.

Diameter in Inches.	Weight in Lb. per Ft.	Breaking Load in Tons of 2,000 Lb.		
		Iron.	Cast Steel.	Plow Steel.
⅜	0.35	3.48	7.00	10.00
₇⁄₁₆	0.44	4.27	9.00	13.00
½	0.60	5.13	12.00	18.00
¾	0.88	8.64	18.00	27.00
⅞	1.20	11.50	25.00	37.00
1	1.58	16.00	33.00	47.00
1⅛	2.00	20.00	42.00	60.00
1¼	2.50	27.00	52.00	75.00
1⅜	3.00	33.00	63.00	90.00
1½	3.65	39.00	77.00	110.00
1⅝	4.10	44.00	86.00	123.00
1¾	5.25	54.00	106.00	157.00
2	6.30	65.00	125.00	189.00

TABLE 47.

IRON AND STEEL WIRE ROPES, 7 WIRES TO THE STRAND

Diameter in Inches.	Weight in Lb. per Ft.	Breaking Load in Tons of 2,000 Lb.		
		Iron.	Cast Steel.	Plow Steel.
½	0.31	2.83	6.00	9.00
⁹⁄₁₆	0.41	4.10	8.00	12.00
⅝	0.57	5.80	11.00	16.00
¹¹⁄₁₆	0.70	7.60	14.00	21.00
¾	0.88	8.80	17.00	25.00
⅞	1.12	12.30	22.00	33.00
1	1.50	16.00	30.00	45.00
1⅛	1.82	20.00	36.00	
1¼	2.28	25.00	44.00	
1⅜	2.77	30.00	52.00	
1½	3.37	36.00	62.00	

2418. The durability of a wire rope depends very largely on the diameter of the drum or sheave over which it is wound. When a rope is bent around a drum, the outer strands must be in tension and the inner strands in compression. This tension, due to bending the rope around the drum, added to the tension to which the rope is already subjected on account of the load, gives the total tension on the part of the rope passing around the drum. This total tension must not exceed the elastic limit of the material composing the wires. *It is just as detrimental to the rope if it be bent partly around the drum or sheave as though bent completely around it.*

2419. From the nature of the rope, it is evident that the diameter of the drum or sheave does not depend upon the diameter of the rope, but only upon the diameter of the wires of which it is made; and we may, therefore, calculate the size of the drum which would be safe for a single wire. It may be stated, as a general rule, *that the diameter of*
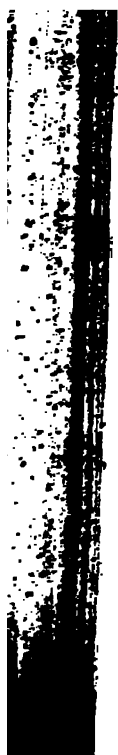
*the drum on which a rope containing 19 wires to the strand
is to be wound should not be less than 60 times the diameter
of the rope, and one on which a rope containing seven wires
to the strand is to be wound should not be less than 100 times
the diameter of the rope.*

It is, of course, preferable to use, if possible, a drum
larger than given by the above rule. Circumstances may
render it necessary to use a drum smaller than given
above, but smaller drums can only be used at the expense
of the rope.

2420. To Splice a Wire Rope.—The length of the
splice depends upon the size of the rope. The larger ropes
require the longer splices. The splice of ropes from $\frac{5}{8}$ inch
to $\frac{7}{8}$ inch in diameter should not be less than 20 feet; from
$\frac{7}{8}$ inch to $1\frac{1}{8}$ inches, 30 feet, and from $1\frac{1}{8}$ inches up, 40 feet.
In Fig. 876 are given a number of illustrations showing the
manner of splicing a wire rope. To splice a wire rope,
proceed as follows:

Suppose it is desired to splice a rope $\frac{5}{8}$ inch in diameter.
The length of the splice for this size rope is, as above given,
20 feet. Tie each end of the rope with a piece of cord at a
distance equal to one-half the length of the splice, or ten
feet back from the end, after which unlay each end as far
as the cord. Then, cut out the hemp center, and bring the
two ends together as close as possible, placing the strands
of the one end between those of the other, as shown in Fig.
876 *A*, in which *k* and *k'* are the cords tied around the rope
ends *M* and *M'* respectively, at a distance of ten feet from
each end. Now remove the cord *k* from the end *M* of the
rope, and unlay any strand, as *a*, and follow it up with the
strand of the other end *M'* of the rope which corresponds
to it, as *a'*; that is, *a* is unwound, leaving a channel in which
a' is wound. About six inches of *a'* are left out, and *a* is
cut off about six inches from the rope, thus leaving two
short ends, as shown at *P* in Fig. 876 *B*, which must be tied
for the present by cords, as shown. The cord *k* should again
be wound around the end *M* of the rope, Fig. 876 *A*, to
prevent the unraveling of the strands; after which remove

the cord k on the other or M' end of the rope, and unlay the strand b; follow it up, as above, with the strand b', leaving the ends out, and tying them down for the present, as before described in the case of strands a and a' (see Q, Fig. 876 B); also, replace the cord k', for the same purpose as stated above. Now again remove the cord k, and unlay the next strand, as c, Fig 876 A, and follow it up with c', stopping, however, this time within four feet of the first set. Continue this operation with the remaining six strands, stopping four feet short of the preceding set each time. We have now the strands in their proper places, with the ends passing each other at intervals of four feet, as shown in Fig. 876 C. These ends must now be disposed of by increasing the size of the rope. Clamp the rope in a vise at the left of the strands a and a', Fig. 876 C, and fasten a clamp to the rope at the right of these strands; then remove the cords tied around the rope which hold these two strands down; after which turn the clamp in the opposite direction to which the rope is twisted, thereby untwisting the rope, as shown in Fig. 876 D. The rope should be untwisted enough to allow its hemp core to be pulled out with a pair of nippers. Cut off twelve inches of the hemp core, six inches at each side from the point of intersection of the strands a and a', and push the ends of the strands in its place, as shown in Fig. 876 D. Then allow the rope to twist up to its natural shape, and remove the clamps. After the rope has been allowed to twist up, the strands tucked in generally bulge out somewhat. This bulging may be reduced by lightly tapping the bulged part of the strands with a wooden mallet, which will force their ends farther into the rope. Proceed in the same manner to tuck in the other ends of the strands.

2421. Preserving Wire Rope.—It is highly essential that wire ropes be oiled, so as to keep them free from rusting, as on this depends in a great measure the life of the ropes. The oil best suited is that which is free from acid, and which will not gum or harden. Oil which hardens greases the outer wires very well, but forms a scale around

them through which the liquid oil can not pass to the inner wires, which therefore go on rusting. Raw linseed-oil has been found to be one of the best oils for this purpose, although a mixture of vegetable tar and raw linseed-oil is extensively used. Either of these may be applied to the rope by saturating the woolly side of a piece of sheepskin. It is well, before applying fresh oil to the rope, to first clean off the old by running the rope between brushes. When the rope is endless and has a continuous motion, as in the case of the endless-rope haulage systems, a different method may be pursued. In this case, the rope may be greased by allowing the oil, which has been placed in a barrel supplied with a cock, to drip on it as it leaves the engine-room on its outward passage. ————

THE ROADBED.

2422. The roadbed for a haulage system is made in practically the same manner as for ordinary railroads. The cross-ties are generally of wood, either 3 in. × 5 in. or 4 in. × 6 in.; they are laid from 18 to 36 inches apart, from center to center, on the solid floor of the mine, and are ballasted. T rails, weighing from 16 to 40 lb. per yard, depending upon the service required, are laid from 30 to 48 inches apart. The wider gauges allow heavier and wider mine-cars to be used, while the narrow gauges permit the cars to run more easily around the curves. The rails laid on slopes should be somewhat heavier than those laid on the level. Gutters should be provided at the lower side of the roadbed, to allow the mine water to flow away, and not destroy the ties. Curves should be made of as large a radius as possible, and it is well never to make the radius less than 25 feet. The gauge of the track should be slightly wider around the curves, and, in case of endless or tail rope haulage, the inside rail should be slightly elevated. In other cases, such as engine-planes, where the empty cars are run by gravity at high velocities, the outer rail should be slightly elevated.

In wire-rope haulage, trouble arises with curves on the roads where the main and tail rope system of haulage is employed. Unless due care is exercised in giving the rails on

a curve the requisite elevation to resist the tendency of the
ropes to pull the ends of the train off the rails at the curve,
the train will frequently be derailed. If, however, the inside
rail is made the higher, the stress due to the inward pull of
the ropes falls on the tread of the wheels and on the upper
sides of the rails. On the other hand, if the rails are equal
in height, the flanges of the inside wheels are pulled against
the inward edge of the inside rail, and then the least jerk
produced at a joint is sufficient to derail the whole train.
When a curve occurs on a highly inclined portion of the road
of a main and tail rope haulage, it is necessary to warn the
engineer to keep the ropes taut when rounding the curves, or
otherwise the high rail may become a source of danger. On
the roads of other haulages, however, the inside rail should
not be raised above the level of the outside one ; for on an
engine-plane and the roads of a self-acting incline, while the
elevation of the inside rail would provide complete security
for the ascending train, it would be a sure means of derailing
a descending one. Therefore, in these cases it is better to
provide a guard-rail set within the rail on the outside of the
curve. For curves on endless-rope haulages, a guard-rail is
required only where the radius of the curve is very short;
otherwise they are not necessary, since the running velocity
is always low.

2423. An arrangement called a covered tube, or con-
duit, which may be used at points where a haulage-road
crosses highways, to enable vehicles to pass over the rope
while it is in motion, is shown in Fig. 877. It consists of
two castings A and A' of the form shown, which constitute
the sides of the tube, or conduit, and which are riveted to
ordinary I beams B. Over these castings are placed two
covers C and C', in the manner shown in the figure. To
strengthen the tube, or conduit, the braces I and I' are riv-
eted to it and to the eye-beams. D and D' are projections
riveted to the castings A and A', respectively, to keep the
covers in their places. The legs of the covers C and C' are
connected by links E and E', forming a toggle-joint; these

links are joined in turn to a rod F, which is connected, through the intervention of a system of levers, to a hand-lever placed at the outside of the track.   By giving an upward movement to the rod F, the links E and E' are forced apart, in consequence of which they cause the covers C and C' to take the position shown by the dotted lines.   It
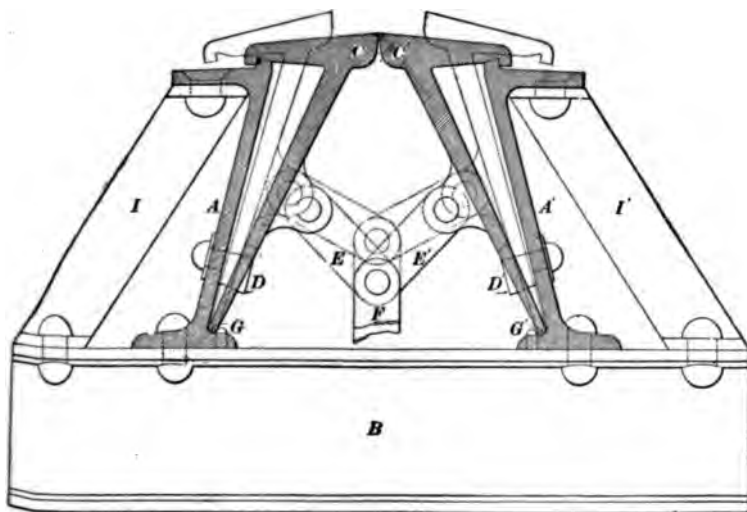
FIG. 877.

is well to construct the conduit in such a manner that vertical lines through the center of gravity of each cover will fall between the points G and G', respectively, when the covers are raised, i. e., in the position shown by the dotted lines, since the covers will then close automatically, it being only necessary to open them.

2424.   Fig. 878 shows the conduit in position.   It will be seen that the length of the conduit is equal to the width of the highway.   Since the cross-ties can not be laid at this point, short rails M and M', having a length equal to that of the conduit, are used.   They are riveted at their ends to chairs P and P', which in turn are riveted to the eye-beams B.   This construction can be built complete in the shop, and put in place as shown in the figure.   The ends of the

rails abutting the short rails M and M' rest on the chairs P and P', and are bolted to them. The covers of the conduit
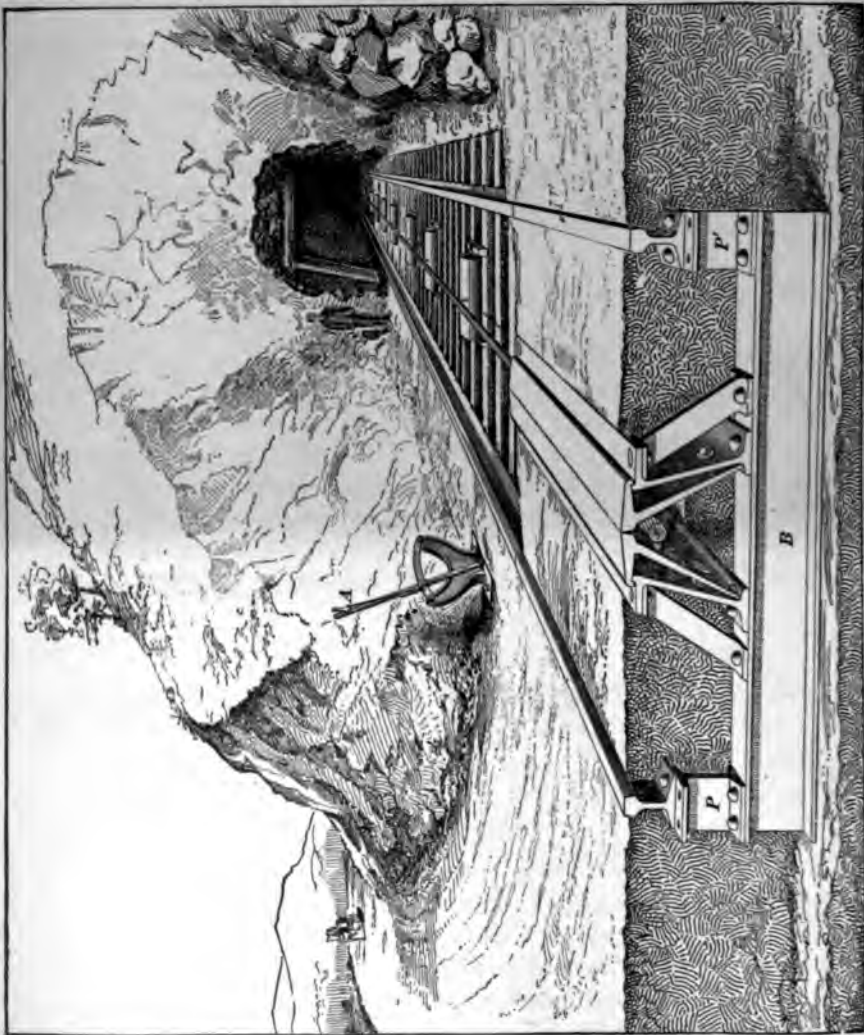
are always closed when no train is to pass, and thus allow vehicles to run across the roadbed. When a car comes along, the hand-lever A located at the side of the track is

pulled to the right, thereby causing the covers of the conduit to open, as before explained, and allowing the train to pass, after which the covers are again closed by the hand-lever.

TRACK MATERIALS.

2425. To find the weight of one mile of two rails in tons **of 2,240** pounds, the weight in pounds per yard of the rail **being** given :

Rule.—*Divide the weight in pounds per yard of the rail by* **7,** *and multiply the quotient by 11.*

EXAMPLE.—How many gross tons of rails, weighing 20 lb. to the yard, are required for one mile of single track ?

SOLUTION.— $\frac{20}{7} \times 11 = 31\frac{3}{7}$ tons. Ans.

2426. To find the number of cross-ties required for one mile of single-track road, the distance between the centers of the cross-ties being given :

Rule.—*Divide 5,280 (the number of feet in a mile) by the distance in feet between the centers of the cross-ties.*

EXAMPLE.—How many cross-ties are required for laying one mile of single-track road, if the cross-ties are laid 21 inches apart from center to center ?

SOLUTION.— 21 inches reduced to feet $= 1\frac{3}{4}$ feet. Then, $5,280 \div 1\frac{3}{4} = 3,017$, nearly. Ans.

2427. The spikes ordinarily used for spiking different **size** rails to the cross-ties, and the average number of spikes **contained** in a 200-pound keg, are given in the following **table** :

TABLE 48.

Size of Spike in Inches.	Weight in Pounds per Yard of Rails Used.	Average Number of Spikes per Keg of 200 Lb.
$4 \times \frac{1}{2}$	25	600
$4\frac{1}{2} \times \frac{1}{2}$	35	525
$5 \times \frac{1}{2}$	35 to 45	448

TRACK-ROLLERS, OR CARRYING-SHEAVES.

2428. To prevent the wire rope of a haulage plant from dragging on the roadbed, or, when the plane is very concave, to prevent it from coming in contact with the roof, **rollers,** or **carrying-sheaves,** are used. These are essential, since otherwise, were the rope permitted to drag on the ground, a greater force would be required to move it, and, besides, its life would be shortened considerably. Rollers, or carrying-sheaves, are made either of wood or cast iron. A form of wooden roller extensively used is shown in Fig. 879, which consists simply of a wooden roller Q having a $\frac{3}{4}$-inch or $\frac{7}{8}$-inch diameter wrought-iron axle R through it. The roller is held in place by the bearings S and S, which are usually nailed to the cross-ties. Ordinarily these bearings are made of wood, and are of such length that they rest on two cross-ties. The bearings should be so located in the center of the track that the rope runs near one end of the roller, so that when the rollers are partly worn they may be turned end for end or shifted sideways, so that the rope will have a new surface to run on. Wooden rollers are also frequently constructed by boring a hole through the center of the roller, and then driving into each end a short length of $\frac{3}{4}$-inch or $\frac{7}{8}$-inch gas-pipe, which acts as the axle of the roller. These rollers are generally made of maple, gum, oak, ash, or beechwood, and vary in diameter from 5 to 8 inches, and in length from 18 to 24 inches. The life of such rollers varies from six to eighteen months.

Fig. 879.

2429. In Figs. 880 and 881 are shown the common forms of cast-iron rollers, or carrying sheaves. In Fig. 880 the roller consists of a cylindrical shell about $\frac{3}{4}$ inch thick, having a flange at each end, as shown, to prevent the rope from being shifted off the rollers sideways. Instead of using

wooden bearings, as already described, iron bearings are used here. These bearings are provided with caps a and a, which can be taken off when the rollers are to be removed. The journals of the shaft are smaller in diameter than the shaft proper, so as to form shoulders on it,

FIG. 880.

in order to prevent the rollers from moving sideways. These rollers are generally made about 6 inches in diameter.
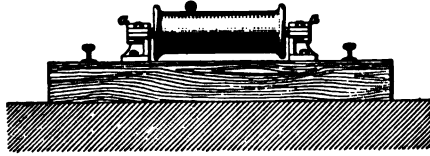
2430. In Fig. 881 is shown another form of cast-iron roller. Here the cylindrical shell of the roller Q is somewhat tapered, and has a small groove in its center, in which the rope runs. The roller is tapered, so that the rope will always slide down to the groove. The shell Q of the roller is fastened to the shaft R by means of the arms and hub S of the roller. The journals t of the shaft are smaller in diameter than the remaining part of it, thus forming shoulders, and preventing the rollers from moving sideways. The bearings U and U are of cast iron, of the form shown in
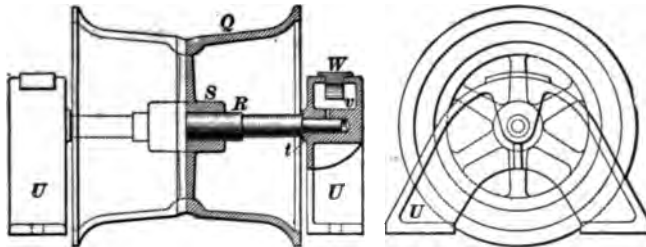
FIG. 881.

the figure, having such a width that they may rest on two cross-ties, to which they are generally bolted. The bearings are provided with a receptacle v, in which oil is poured through the top (by removing the cap W) for lubricating the journals. These rollers, or carrying-sheaves, are generally made 10 or 12 inches in diameter.

UNDERGROUND HAULAGE-ENGINES.

2431. There are two kinds of hauling-engines in use for mine haulage, each specially fitted for different classes of work. For example, engines on the first motion, or those connected directly to the drum-shaft, are best adapted for long runs and level roads; on the other hand, engines on the second motion do the best work on short runs, or in hauling loaded trains up steep inclines pitching from the shafts. The engine and drums illustrated by Fig. 882 furnish an example of an engine on the second motion. Here the drums are seen mounted on a heavy shaft, and are geared
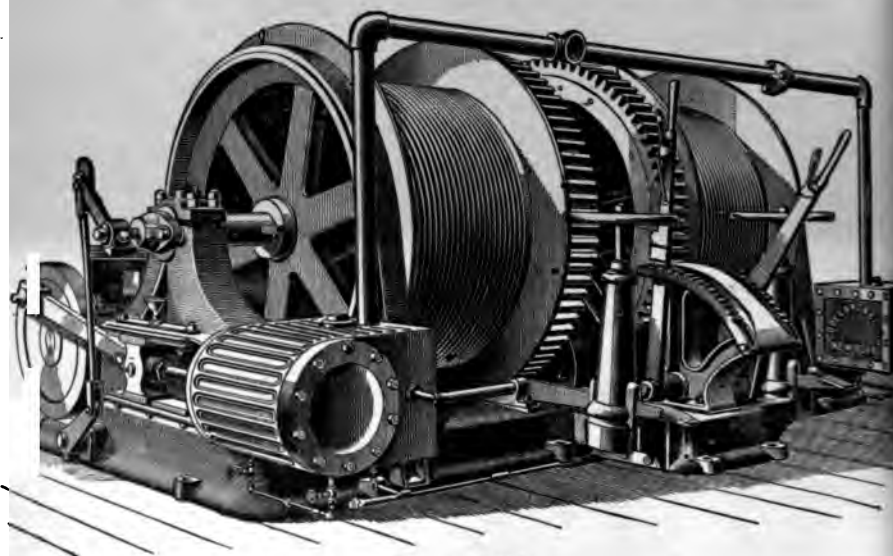
FIG. 882.

to the engine by massive spur-gears and pinions. This arrangement is designed for main and tail rope haulage; consequently, the drums are both made to run loose on the shaft when not required for hauling. When the main-rope drum is required to haul out a train of cars to the shaft, it is secured to the spur-wheel with a friction-clutch, or other-

wise the pinion-wheels on the engine-shaft are made to slide
in and out of gear. The pinions are made to lock onto the
engine-shaft with strong keys and slots, and so connected
that, as one pinion is put into gear, the other is thrown out.
These changes are made by a lever and slot and fork ar-
rangement, which is controlled by the engineer. After a
train has been hauled out with the main rope, the main-rope
drum is thrown out of gear and made to run loose, so that,
during the hauling in with the tail-rope drum, the main rope
is hauled in with the train; *vice versa*, to haul out, the main-
rope drum is put into gear, and, consequently, the tail-rope
drum is thrown out of gear when the tail-rope is hauled out
with outgoing trains of loaded cars.

The friction-clutch connection is common where the en-
gine is on the first motion, for then the drums are mounted
on the engine-shaft. The braking of the main and tail rope
drums requires special care and attention on the part of the
engineer, for otherwise he may hold the loose running drum
too tight, and in that case unduly strain the rope engaged
for the time in hauling, and at the same time waste the
available energy of the engine and reduce the velocity of
the train. On the other hand, if the running-off rope is not
kept comparatively tight, it may overrun and kink and be
destroyed. Not only should the engineer be a man of pru-
dence, but, for safety and economy, the brake arrangement
should be so constructed that it will secure in action the
following good points:

(1) It must be strong and reliable; (2) the friction gen-
erated must be sufficient to stop and hold the train secure
on any part of the road; (3) the brake must be actuated
by a compound lever, so as to exert a great stress with
the power of an ordinary man's arm; (4) the brake-handle
must be fixed in a position so near the throttle-valve of the
engine that the engineer can turn from one to the other
without changing his position. Fig. 883 shows a brake and
lever arrangement that possesses all the good points just
referred to.
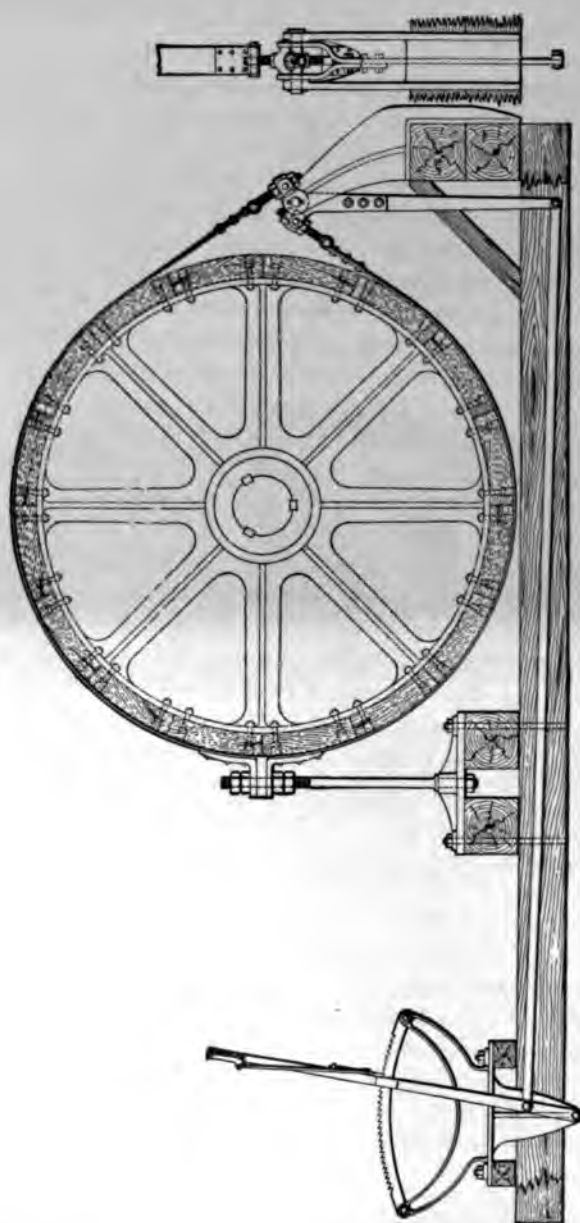
FIG. 88.

UNDERGROUND HAULAGE BY ENDLESS ROPE.

2432. Haulage-drums can not be used for an endless-rope haulage, as in this case two ropes must run continuously in opposite directions; therefore, it is out of the question to even think of using reels. Reels, or drums, however, can not be disposed of without substituting wheels that not only coil on but coil off at the same time, and yet seize the rope for hauling as securely as a drum would do. Now it is clear that a single-grooved pulley could not seize the rope sufficiently for hauling against a resistance greater than the friction due to a rope slipping in the groove whose length equal to half the circumference of a wheel. Again, it is not possible to make two or more turns of the rope round a wheel, and haul continuously by this means, because the rope would all run on at one side, and, therefore, after a turn or two, would coil on itself. It is true that fleet-wheels have been contrived by which the coils are made to continuously slip and surge from one side of the tread of the wheel towards its center; but the surging and slipping on such a wheel generates a heavy strain on the rope, and this, added to the wear by the friction due to slipping, soon destroys a good rope, and renders this system of gripping the rope a practical failure.

2433. By a principle in mechanics, two grooved-heels can be made to secure sufficient gripping force, and run the rope on and off without injury from surging or any other cause. To understand the mode of action, suppose that a rope is made to perform the circuits of eight semicircular grooves on two four-grooved pulleys, as shown in plan at the top of Fig. 884. Now let us follow a point of the rope throughout its journey, from the time it enters the first groove to the moment of leaving the last one. The rope runs half round the first wheel, and then runs off to the first groove of the second wheel. It advances to the second groove of the first wheel, and then passes half round and runs off to the second groove of the second wheel. The point passes round the second groove of the second wheel,
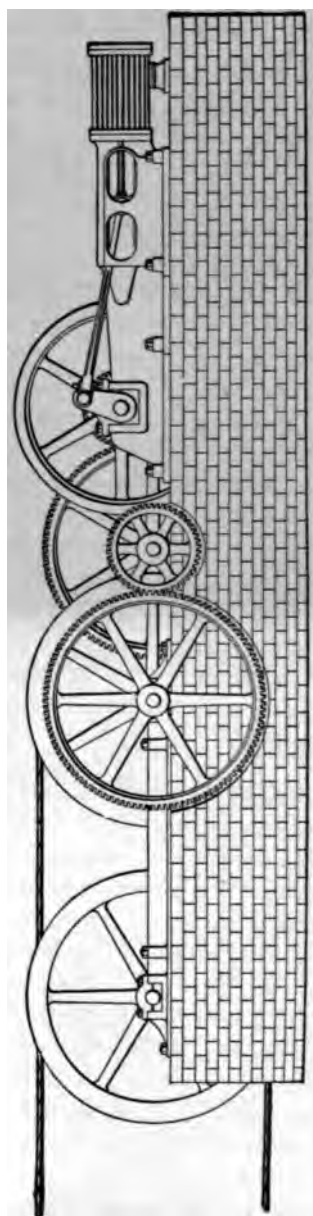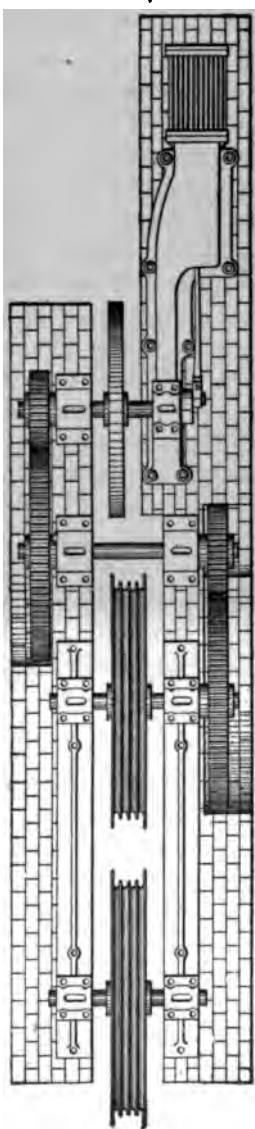
Fig. 664.

continues its journey to the third groove of the first wheel, from that to the third groove of the second wheel, and so on. It will be found that the rope leaves the fourth groove of the second wheel, and pursues its journey in the line of the haulage. The eight semicircles are equal to four complete turns round one drum, and the grip due to the four turns is nearly equal to the breaking strain of the rope. The two grooved wheels solve the problem of providing a secure grip for a haulage that is equal to that provided by a drum for hauling on an ordinary engine-plane. In the lower portion of the figure, the engine is seen to be on the third motion; this means that a small engine is made to run at a high velocity and thus develop as much power as could be developed by a large engine on the first motion.

2434. In hauling and hoisting, the position of the train or the cage can not be seen by the engineer, and, as safety is required, it must be secured by some mechanical contrivance for indicating the position of the train or the cage. It is important that the engineer should know how to regulate his supply of steam, and when to apply the brake, in case the road over which the train is running is pitching. He also should know to a foot where to stop the train at the different stations on the haulage-roads. Again, the engineer at the hoisting-engine must know the exact position of the ascending cage, so that he can run at a maximum speed without fear, and stop the cage within an inch of the level of the landing-stage. From this it can be seen that the shaft or haulage-road indicator is indispensable. Fig. 885 is an indicator for a haulage-road. The velocity ratios of the train and the indicator-weight are so proportioned that the latter only moves over a space of one foot while the train on the haulage-road is running a distance of 300 feet. The velocity ratios are 1 for the indicator to 300 for the train. This reduction in the velocity of the indicator-weight is secured as follows: A worm on the end of the drum-shaft runs in the teeth of a worm-wheel, and this wheel makes only 1 revolution for 30 turns of the drum. Again, the

FIG. 885.
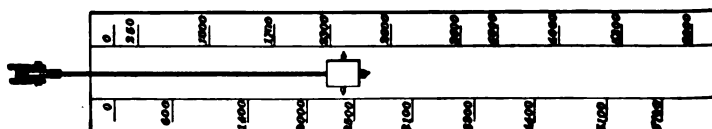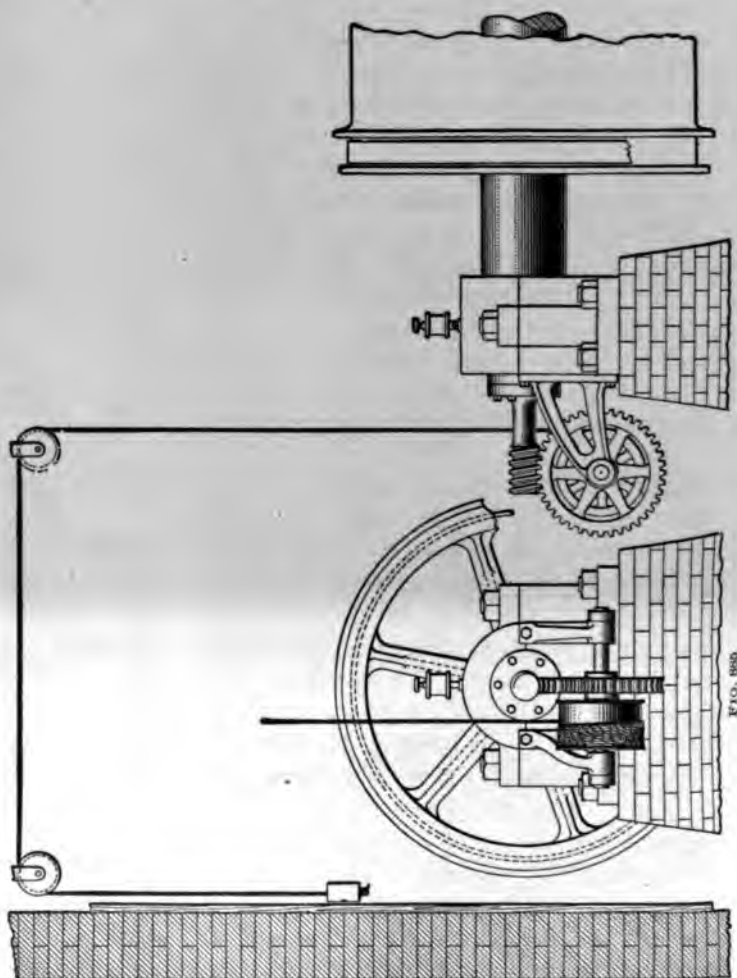
diameter of the barrel for winding on the indicator-cord is only one-tenth of the mean diameter of the drum for the haulage-rope, and, therefore, $30 \times 10 = 300$, the velocity ratio of the haulage-rope. The worm-wheel and the drum for the indicator-cord are plainly shown in the figure.

HAULAGE BY MINE LOCOMOTIVES.

2435. There are three types of mine locomotives, each of which has been successfully used in mine haulage; and there will shortly be on the market a fourth type, which promises to be, in a measure, a competitor of the other three. The first three types are: steam locomotives, compressed-air locomotives, and electric locomotives. The fourth is a locomotive operated by a gasoline-engine.

2436. The steam locomotives used in mining work are practically small locomotives of the ordinary type; they usually have two driving-wheels, and sometimes three, on each side; the water-tank is set over the boiler, and the smokestack is shortened so that it does not extend above the top of the boiler; in short, the smokestack, top of boiler, and top of small cab are on the same level, and low enough to permit the locomotive to enter the mine-passages. They are usually made with a short wheel-base, so as to enable them to run around sharp curves.

2437. While these small locomotives are convenient, and very efficient around mines for outside haulage, they
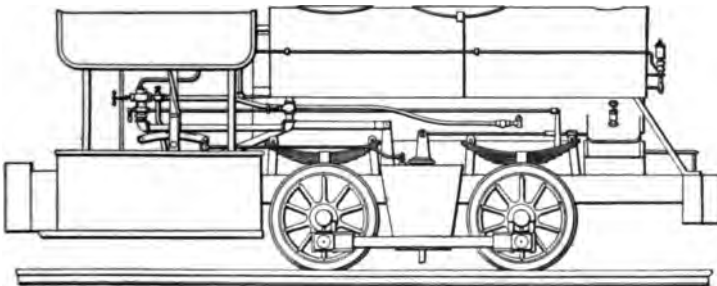
FIG. 886.

should not be used for inside haulage if their use for such

haulage can be avoided, because the exhaust from the smokestack vitiates the mine air, and the fire under the boiler is extremely dangerous in any mine in which explosive gas is generated. When steam locomotives are of necessity used in mine haulage, the haulage-road should always be the return-air course; they should never be used in mines in which explosive gas is found. Fig. 886 shows a view of a typical steam mine locomotive. Where such locomotives are used, the track should be as smooth and well ballasted as possible.

2438. The compressed-air locomotives used in mine haulage have their working parts built very similar to those of steam locomotives, the main difference being that the boiler and fire-box are replaced by one or two compressed-air tanks, which are usually charged with compressed air at suitable intervals along the haulage-road. The compressed-air locomotive can be operated with safety in any part of a mine where grades will permit, and is not dangerous in the event of the mine producing explosive gases. However, the installation of a compressed-air locomotive is much more expensive than that of the steam locomotive.

2439. The compressed-air locomotive requires a stationary plant on the surface, consisting of an air-compressor of proper design for the quantity and pressure of air necessary to run the locomotive; in addition, unless water-power is available to run the compressor, a boiler to generate steam for that purpose is required. The air from the air-compressor must be conveyed through a pipe-line to the point or points where it is convenient to charge the locomotive. Each charging-station consists of a heavy valve with a metallic flexible coupling, bleeder-valve, and screw-joint, which couples to a similar screw-joint on the locomotive. While it is entirely practicable to charge a compressed-air locomotive direct from the compressor, without any intermediate storage-reservoir, it is not economical unless a number of locomotives are used, so that, while one locomotive is being charged, the rest are making trips.

2440. The best and cheapest method is to operate the compressor at a nearly uniform speed, pumping continuously in the storage-reservoir from which the locomotive is charged. If the compressor is only run while the locomotive is charging, the wear and tear on the compressor and boiler are greater, steam and fuel are wasted, an unnecessarily large compressor is required, and the locomotive is held idle longer than desirable. One or more stationary tanks may be used for storage-reservoirs, but a pipe-line is preferable, because it is handier, and provides opportunity for a number of charging-stations at different points, and also makes it easy to charge different locomotives in different places or on different levels of the mine, besides conveying the air wherever desired.

2441. The whole operation of stopping the locomotive, connecting it to the charging-station, charging it to the required pressure, and disconnecting, and starting on its trip, can usually be performed in less than one and one-half minutes.

2442. The pressure and cubic capacity of the pipe-line or tank should be so proportioned that, when the locomotive returning for a new charge is connected, the pressure of the locomotive and of the pipe-line equalize almost instantly to the required pressure. It is, therefore, necessary to carry a higher pressure in the pipe-line than is needed for the locomotive. The necessary pressure of air and capacity of the pipe-line can very easily be determined. For instance, if the locomotive-tanks have 100 cu. ft. capacity, the charging pressure needed for the locomotives is 500 lb., and the average pressure remaining in the locomotive, when returning for the new charge, is 50 lb., a pipe-line of 300 cu. ft. capacity and 650 lb. pressure will, when connected with the locomotive, charge it instantly to 500 lb. This is shown by the following calculation:

The pressure in the locomotive-tank is 50 lb. and that in the pipe-line is 650 lb.; the volume of the tank is 100 cu. ft. and that of the pipe-line is 300 cu. ft. The resulting

pressure, when the two are connected, may be obtained from formula **19** in Gases Met With in Mines.   Thus,

$$P = \frac{p\,v + p_1 v_1}{V} = \frac{50 \times 100 + 650 \times 300}{100 + 300} = 500 \text{ lb. per sq. in.,}$$

the pressure needed for the locomotive.   Practically the same result would be reached by a pipe-line of 400 cu. ft. at 610 lb. pressure, or 250 cu. ft. at 680 lb. pressure.

2443.   In actual practice, for the sake of economy, the air-compressor may be regulated so as to charge the pipe-line to a lower pressure, whenever it may be desired to run the mine at less than its full output.   The compressor can be made automatic, and may be set for any required maximum pressure, and to retard its speed as this limit is approached. Relief-valves should be used for both the pipe-line and for the locomotive, as a protection against over-charging.   In some cases, where the pipe-line extends beyond the charging-station quite a distance, a less pipe-line pressure may be used; and after the locomotive has been connected with the pipe-line, and the pressure has been equalized, that part of the pipe-line beyond the charging-station may be shut off by a valve, and the locomotive may continue, connected by the remaining part of the pipe-line to the compressor, until the required pressure is reached, when the locomotive can be disconnected and the valve opened.   There is no difficulty in obtaining and laying pipe strong enough to stand 500 to 1,000 lb. pressure to the square inch, and this form of reservoir costs very nearly the same as equivalent storage in tanks.   Four to six inches diameter is usually the best size for pipe-line.   Air may be conveyed several miles through a pipe of as small diameter as two inches with scarcely appreciable loss of pressure.   For a pressure of 500 to 600 lb. or more, a three-stage compressor is generally preferred. If a low-pressure compressed-air system is also used for operating mine machinery, such as drills, pumps, hoists, etc., it is often economical to supply the locomotive with air from the low-pressure system, at, say, 60 to 100 lb. pressure; in this case, a special auxiliary two-stage compressor is used.

For extra light work, short trips, and easy grades, a pressure of 80 to 100 lb. may be sufficient for the locomotive.

2444. Air-compressors are now so constructed that they may be used during the daytime to furnish high-pressure air from 400 to 800 lb. for operating the locomotives, and during the night to furnish air at low pressure—60 to 100 lb.—for operating drills, etc. This arrangement is convenient for night-work, whether it consists in driving entries or in getting mine faces ready for the miners.

2445. As stated before, the mechanical construction and practical operation of the compressed-air locomotive is similar to that of the steam locomotive. Air-tanks take the place of the boiler, and the air is applied and used in much the same manner as steam; the details, however, are modified to secure the most economical and convenient use of the air. The main air-tanks should be made of a special quality of heavy steel plates, with heavy butt-riveted welt-strip horizontal seams, and with both heads flanged convex. The front head should be provided with a manhole. The usual range of charging pressure for compressed-air locomotives is 500 to 800 lb., although heavier pressure can be used in case of long hauls and heavy work, with but one charge of air.

2446. Experience has shown, in well-constructed air pipe-lines, that slight changes in temperature cause greater
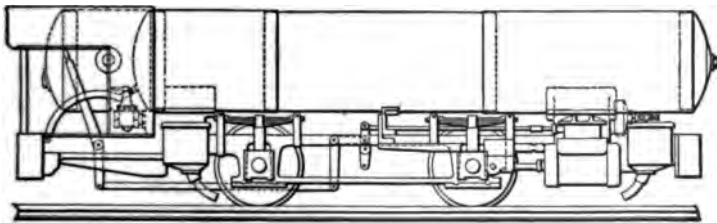
FIG. 887.

variations in pressure than are produced by leakage. A properly constructed pipe-line, with heavy connections, carefully fitted, and supplied with special valves, will lose very little, if any, through leakage. While the first cost of a

compressed-air haulage plant is greater than that of a steam locomotive, the cost of maintenance is less; the cost of operating, however, exclusive of repairs, is slightly more when but one locomotive is used. When several compressed-air locomotives are used, the difference in cost of mere operation becomes less, until it reaches a point at which it is really cheaper than the operation of the same number of steam locomotives. Figure 887 is a side view of a modern compressed-air mine locomotive.

2447. Electric mine locomotives consist essentially of a heavy truck, the wheels of which are rotated by a mechanism operated by an electric motor. The operation of the electric mine locomotive is very similar to that of an electric trolley-car; in fact, the power to run the locomotive is conveyed along the haulage-road through a conducting wire on which a trolley connected with the locomotive runs.

2448. The electric locomotive possesses one great advantage over the steam locomotive: there is no exhaust to vitiate the air of the mine. It is, therefore, an excellent haulage-machine for use in mines free from explosive gas. Its use requires the erection of a plant for generating electricity on the surface. The power thus generated is very convenient, and the current carried along the conducting wires can be utilized for many purposes in mines. For instance, coal-cutters, pumps, drilling-machines, hoists, etc., etc., can be operated by it. This gives electric haulage plants advantages which have resulted in their adoption in many mines. Actual results prove them very efficient and economical in operation.

2449. Gasoline mining locomotives have not as yet been used, although the successful utilization of gasoline-engines for other purposes makes it evident that one manufacturer's claim of his ability to produce a gasoline locomotive is well founded. Such a locomotive, however, will be open to one of the objections for mine use that applies to a steam locomotive, viz., the vitiation of the mine air by the

products of the combustion of the gasoline vapor in the cylinders, which must pass out through the exhaust.

2450. All the types of mine locomotives mentioned require for successful operation that the grades of the roads over which they run be comparatively light; in this respect, they are less advantageous than rope haulage. Again, as stated before, with mine locomotives the haulage is performed in a more intermittent manner than by rope haulage, more especially than by the endless-rope system. In the construction of mine-tracks for locomotive haulage, the same general rules apply as in the construction of tracks for rope haulage, the only difference being that, when locomotives of any type are used, the outer rail of all curves should be slightly elevated. The same rule regarding the size of curves applies in the case of locomotive haulage as in rope haulage, namely, that they should be of as large radius as possible, though it must be said that well-designed mine locomotives with a short wheel-base will run and pull trips around sharp curves better than any type of rope haulage.

A SERIES

OF

QUESTIONS AND EXAMPLES

Relating to the Subjects
Treated of in This Volume.

It will be noticed that the questions and examples contained in the following pages are divided into sections corresponding to the sections of the text of the preceding pages, and that each section has a headline which is the same as the headline of the section to which the questions refer.   No attempt should be made to answer any questions or to work any examples until the corresponding part of the text has been carefully studied.
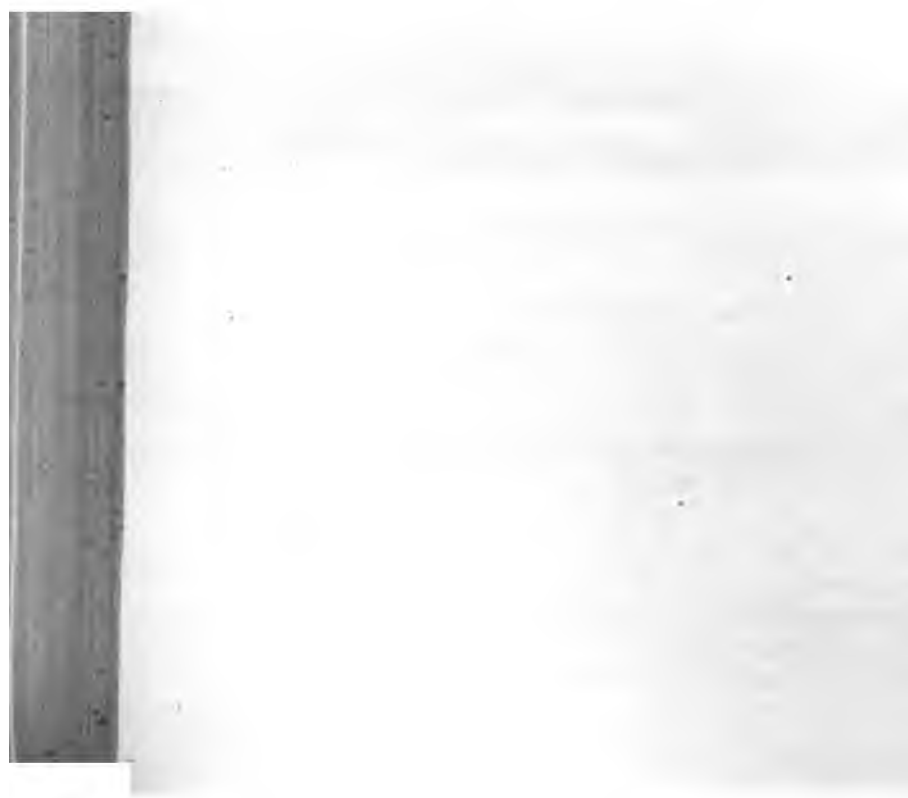
MECHANICS.

(PART 1.)

EXAMINATION QUESTIONS.

(1) (*a*) What is a molecule ?   (*b*) What is an atom ?

(2) The number of teeth in a spur-gear is 50 and the pitch is 1½ inches; (*a*) what is the pitch diameter ?   (*b*) What is the outside diameter ?   Ans. $\begin{cases} (a) \ 23.87''. \\ (b) \ 24.77''. \end{cases}$

(3) What pressure can be exerted by a force of 24 pounds on a half-inch screw which has 13 threads per inch, the distance from the center of the screw to the point on the handle where the force is applied being 11 inches ?

Ans. 21,563.94 lb.

(4) A ball weighing 5 pounds revolves in a circle whose radius is 32 inches, at the rate of 350 R. P. M.; what is the pull on the support caused by the ball ?   Ans. 555⅓ lb.

(5) A body weighing 2 pounds has a velocity of 600 feet per second; what is its kinetic energy ?   Ans. 11,194 ft.-lb.

(6) What should be the width of a double leather belt to transmit 150 horsepower, when the belt has a velocity of 3,000 feet per minute, and has 7 feet of its length in contact with the smaller pulley, whose diameter is 63 inches ?   Give width to nearest half inch.   Ans. 29.5 in.

(7) (*a*) What are the three states of matter ?   Name (*b*) some of the general properties of matter; (*c*) some of the specific properties.

§ 16

(8) What is meant by *center of gravity?*

(9) (*a*) Why is crowning usually given to the face of a pulley? (*b*) Why should high-speed pulleys be balanced?

(10) At what speed must the engine run when the diameter of the band-wheel is 13 feet and of the main pulley 91 inches, if the speed of the main shaft is to be 108 R. P. M.?

Ans. 63 R. P. M.

(11) What do you understand by *specific gravity?*

(12) What should be the width of a single leather belt to transmit 2¼ horsepower when the belt has a velocity of 2,000 feet per minute? The diameter of the smaller pulley is 14 inches, and the belt has 18 inches of its length in contact with it.

Ans. 1¼ inch.

(13) What is meant (*a*) by inertia? (*b*) by weight? (*c*) How is weight measured?

(14) The speed of a certain belt is 3,000 feet per minute; if it drives a 48-inch pulley, how long will it take the pulley to make 100 revolutions?

Ans. 25.13 sec., nearly.

(15) Find the point of suspension of a rectangular cast-iron lever 4 feet 6 inches long, 2 inches deep, and ¾ inch thick, having weights 47 and 71 pounds hung from each end, in order that there may be equilibrium. Take the weight of a cubic inch of cast iron as .261 pound.

Ans. $\begin{cases} \text{Short arm} = 22.343 \text{ in.} \\ \text{Long arm} = 31.657 \text{ in.} \end{cases}$

(16) A cubic foot of a certain kind of wood weighs 51 pounds; what is its specific gravity?

Ans. .816.

(17) What is (*a*) motion? (*b*) velocity? (*c*) rest? (*d*) Can a body be in motion with respect to one object and at rest with respect to another? Explain fully.

(18) (*a*) What is force? (*b*) Name several kinds of forces.

(19) Find by measurement the center of gravity of a triangle whose sides are 4 inches, 5 inches, and 6 inches long.

Ans. 1$\frac{3}{31}$ inches from 6-inch side.

(20)   What horsepower can be safely transmitted by a gear whose pitch is 1.57 inches, pitch diameter is 30 inches, and which makes 100 revolutions per minute ?

Ans. 19.36 H. P.

(21)   (*a*) What is uniform motion ?   (*b*) What is variable motion ?   (*c*) If a body moves 10 feet the first second, 12 feet the second second, 15 feet the third second, etc., is its motion uniform or variable, and why ?

(22)   In a train of gears used to raise a weight of 6,000 pounds in a manner similar to that shown in Fig. 612, the diameters of the drivers and belt pulley are 18 inches, 12 inches, 15 inches, and 12 inches, and of the pinions and drum, 6 inches, 5 inches, 8 inches, and 3 inches.   What force must be applied to the belt to raise the weight, if 20% of the total force is lost through friction ?   Ans. 138⅜ lb.

(23)   The pitch diameter of a gear is 24.16 inches, and the number of teeth is 38; what is the pitch ?

Ans. 1.9974 in.

(24)   It is required to raise a load of 1,890 pounds by means of a block and tackle which has four fixed and four movable pulleys; what force is required to be applied to the free end of the rope ?   Ans. 236¼ lb.

(25)   A piece of lead is ¼ inch in diameter and 10 inches long; how much does it weigh ?   Ans. 12.91 oz.

(26)   It is required to raise a weight of 1,500 pounds by means of a lever like that shown in Fig. 596.   The length of the lever is 4 feet, and the distance from the fulcrum to the weight is 4 inches; what force will it be necessary to apply ?   .   Ans. 136$\frac{4}{11}$ lb.

(27)   Had the lever in the above example been like that shown in Fig. 597, what force would have been required ?

Ans. 125 lb.

(28)   What is (*a*) a spur-gear ?   (*b*) a miter-gear ?   (*c*) a bevel-gear ?

(29)   The length of an inclined plane is 400 feet and the height is 45 feet.   What force acting parallel to the

plane will be required to pull up the plane a weight of 4,000 pounds?   Ans. 450 lb.

(30) The diameters of two pulleys are 14 inches and 18 inches, and the distance between their centers is 14 feet; what must be the length of a belt to drive these pulleys?
                                                          Ans. 32 ft. 4 in.

(31) What is (*a*) a rack? (*b*) a worm-wheel? (*c*) a worm?

(32) (*a*) What distinguishes epicycloidal teeth from the involute teeth? (*b*) Name two advantages which the latter possess over the former.

(33) An inclined plane has a length of 1,200 feet and a height of 125 feet. It is required to pull a load of 50,000 pounds up this plane. A block and tackle having 6 fixed and 6 movable pulleys is stationed at the top of the plane, and the weight end of the rope is attached to the load. If the rope which connects the block to the load is parallel to the plane, what force will it be necessary to exert on the free end of the rope to pull up the load, no allowance being made for friction?   Ans. 434 lb.

(34) What do you understand (*a*) by centrifugal force? (*b*) by centripetal force?

(35) Why is it difficult to jump from a rowboat?

(36) A compound lever, similar to the one shown in Fig. 602, is required to lift a weight of 1,250 pounds. The lengths of the power-arms $P F$ are 30 inches, 20 inches, 10 inches, and 15 inches, respectively, and the lengths of the weight-arms $W F$ are 6 inches, 5 inches, 4 inches, and 7 inches; what force will be required?   Ans. 11⅔ lb.

(37) How is the diameter of a gear measured?

(38) How much work can be done by 20 cubic feet of water falling from a height of 50 feet?   Ans. 62,500 ft.-lb.

(39) It is required to raise a weight of 18,000 pounds by means of a screw having 3 threads per inch; if the length of

the handle is 15 inches, and there is a loss of 10,000 pounds, due to friction, etc., what force will it be necessary to apply to the handle ?   Ans. 99 lb., nearly.

(40)   If the distance between the center line of the handle and the axis of the drum shown in Fig. 604 is 14½ inches, and the diameter of the drum is 5 inches, what load will a force of 30 pounds exerted on the handle raise ?

Ans. 174 lb.

MECHANICS.

(PART 2.)

EXAMINATION QUESTIONS.

(1) What is meant by the expression, *the resultant of several forces?*

(2) If in Fig. 631 the tension in the rope is 3¾ tons, and the angle at *d* between the directions of the two parts of the rope is 30°, what is the total load on the shaft of the head-wheel?

(3) What do you understand (*a*) by tensile strength of a material? (*b*) by working stress?

(4) A close-link wrought-iron chain is made from ⅝-inch iron; what is the greatest safe load that it will carry?

Ans. 1,687.5 lb.

(5) What is the allowable working load for a steel-wire rope 5¼ inches in circumference? Ans. 27,562.5 lb.

(6) If a line 5 inches long represents a force of 20 pounds, (*a*) how long must the line be to represent a force of 1 pound? (*b*) of 6¼ lb?

(7) What is (*a*) cold-rolled shafting? (*b*) bright shafting? (*c*) black shafting?

(8) Find the resultant of the forces acting in Fig. I—all acting towards the same point?

FIG. I.

§ 17
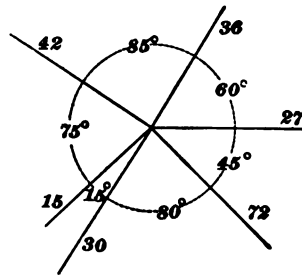
(9) What load can be safely sustained by a round wooden pillar, 8 inches in diameter and 10 feet long, having both ends flat ?   Ans. 13¼ tons.

(10) What are the components of a force ?

(11) What should be the least diameter of a wrought-iron bolt that is to resist a sudden pull of 12,000 pounds ?
Ans. 1.74+ in.

(12) A white-pine beam supported at both ends has a rectangular cross-section 8 inches wide by 10 inches deep; if the beam is 28 feet long, what total uniform load will it support in safety ?   Ans. 6,857¼ lb.

(13) What horsepower can a 10-inch wrought-iron crank-shaft transmit when running at 200 revolutions per minute?
Ans. 2,857¼ H. P.

(14) A force of 87 pounds acts at an angle of 23° to the horizontal; what are its horizontal and vertical components? Find, first, graphically, by the method of triangle of forces, and, second, by trigonometry.   Ans. $\begin{cases} 33.994 \text{ lb.} \\ 80.084 \text{ lb.} \end{cases}$

(15) What is the greatest safe load that may be applied to a stud-link wrought-iron chain, if the diameter of the iron from which the link is made is ¼ inch ?   Ans. 4,500 lb.

(16) It is desired to haul loads up to 14,000 pounds by means of an iron-wire rope; what should be its circumference ?   Ans. 4.83 in., nearly.

(17) Two forces act upon a body at a common point—one with a force of 75 pounds, and the other with a force of 40 pounds; if the angle between them is 60°, and both forces act towards the body, what is the value of the resultant ? Solve by the method of triangle of forces and parallelogram of forces, and mark the direction of the resultant.
Ans. 101+ lb.

(18) In the last example, if one force (the one of 75 pounds) acts away from the body, and the other towards it, what is the resultant ? Solve by the method of triangle of forces

and parallelogram of forces, and mark the direction of the . resultant.   Ans. 65 lb.

(19) If two forces, of 27 pounds and 46 pounds, respectively, act in exactly opposite directions upon a body, what is the resultant ?

(20) A bar of steel having a cross-section of $1\frac{3}{4}$ inches by 3 inches is subjected to a tensile stress; if the stress is suddenly applied, what is the greatest load that it will safely carry ?   Ans. 31,500 lb.

(21) In laying out an engine-plane, it was found necessary to lead the rope around two guiding-sheaves, as shown
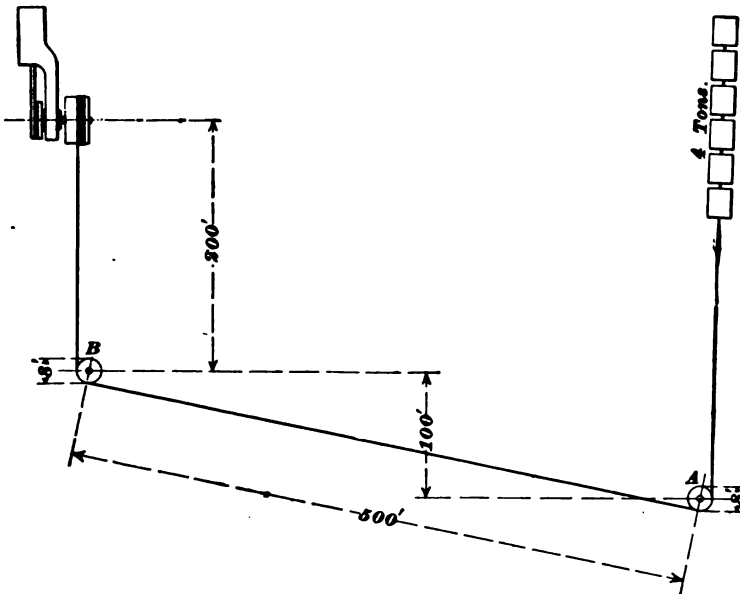
FIG. II.

in Fig. II. The portion of the rope between the car and the sheave A is parallel to the portion of the rope led from the engine to the sheave B. The locations of the sheaves are found from the dimensions given. The resistance due to the cars and coal—that is, the tension in the rope—is

4 tons. What is the greatest pressure on the shaft of each sheave? Solve graphically by means of the parallelogram of forces.   Ans. $\begin{cases} \text{Pressure on sheave } A, \text{ 12,400 lb., nearly.} \\ \text{Pressure on sheave } B, \text{ 10,125 lb., nearly.} \end{cases}$

(22) What is (*a*) a stress? (*b*) a strain? (*c*) a unit stress?

(23) A steel-wire rope is used to haul cars up an inclined plane; the greatest stress in the rope is 8,000 pounds; what should its circumference be?   Ans. 2.83 in.

(24) What uniform load can be safely sustained by a steel beam 20 feet long, 2 inches wide, and 6 inches deep?
                                Ans. 4,608 lb.

(25) What is (*a*) elasticity? (*b*) elastic limit? (*c*) What is meant by set?

(26) What is the allowable working load for an iron-wire rope 6 inches in circumference?   Ans. 21,600 lb.

(27) What force is required to shear a wrought-iron strip 4 feet long and $\frac{1}{4}$ inch thick?   Ans. 960,000 lb.

(28) A 7-inch wrought-iron crank-shaft is to transmit 200 horsepower; how many revolutions per minute must it make?   Ans. 40.8 rev., nearly.

(29) An iron-wire rope 4 inches in circumference is used for hoisting; what is the greatest load that the rope will sustain with safety?   Ans. 9,600 lb.

(30) A cast-iron rectangular cantilever beam, having a cross-section of $1\frac{1}{2}$ inches wide by $2\frac{1}{2}$ inches deep, is 4 feet 8 inches long; how great a weight will the beam sustain at its end?   Ans. 201 lb., nearly.

(31) What horsepower will a $2\frac{7}{16}$-inch steel shaft transmit when running at 120 revolutions per minute, there being pulleys between bearings?   Ans. 20,445 H. P.

(32) What safe steady load can be sustained by a $1\frac{1}{2}$-inch round wrought-iron bar, the load producing a tensile stress?
                              Ans. 21,205.2 lb.

(33)   What load will a hollow cast-iron pillar support with safety, if the pillar is 20 feet long, outside diameter 14 inches, inside diameter 11¼ inches, and both ends are fixed?

<div align="right">Ans. 219.24 tons.</div>

(34)   What force is required to punch a hole 1¼ inches in diameter through a ¾-inch steel plate?   Ans. 212,058 lb.

(35)   A weight of 325 pounds rests upon a smooth inclined plane, as shown in Fig. 636. If the angle of the plane is 15°, (*a*) what is the perpendicular pressure against it? (*b*) What force would it be necessary to exert parallel to the plane to keep it from sliding downwards, there being no friction? Solve by trigonometry, and also graphically by the method of the triangle of forces.   Ans. $\begin{cases} (a) \ 313.93 \text{ lb.} \\ (b) \ 84.12 \text{ lb.} \end{cases}$

STEAM AND STEAM BOILERS.

EXAMINATION QUESTIONS.

(1188) (*a*) What is heat ? (*b*) Suppose a closed vessel containing air is placed in a furnace; describe the effect of the heat upon the molecules of the air. (*c*) If the vessel is so arranged that the air can not escape or expand, will the pressure of the air increase as it is heated ? (*d*) Why ?

(1189) (*a*) What is temperature ? (*b*) Describe the thermometer. (*c*) Of what is temperature a measure ?

(1190) A bar of iron weighing 2½ pounds has a temperature of 460°. Does the bar contain more or less heat than 10 pounds of water at 60° ?

(1191) (*a*) What are some of the effects of heat ? (*b*) Give some practical illustrations of the expansion of bodies by heat.

(1192) (*a*) What is a B. T. U. ? (*b*) What is latent heat ? (*c*) What is sensible heat ? (*d*) What is meant by the specific heat of a substance ?

(1193) A pound of ice at 16° is heated until it finally is changed to steam at atmospheric pressure. (*a*) Describe the action of the heat upon the ice and water. (*b*) How many B. T. U. are required for the operation ? (*c*) What part of the heat applied is sensible heat ? (*d*) What part is latent heat ?

(1194) (*a*) What is the mechanical equivalent of heat ? (*b*) Give examples of heat changed to work, and *vice versa*. (*c*) How many foot-pounds of work are equivalent to 30½ B. T. U.? Ans. (*c*) 23,729 ft.-lb.

(1195) Assuming 20% of the heat to be utilized, how

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many heat-units per hour are required to run an engine developing 35 horsepower?   Ans. 445,372.5 B. T. U.

(1196)   In a power plant, the engine extracts 8% of the heat produced by the combustion of the coal.   Assuming that the combustion of the coal produces 14,000 B. T. U. per pound, how many pounds of coal per H. P. per hour are used by the engine?   Ans. 2.27 lb.

(1197)   How many B. T. U. are required to raise the temperature of $22\frac{1}{2}$ pounds of sulphur from 44° to 68°?
        Ans. 109.4 B. T. U.

(1198)   (a) What is the latent heat of fusion of ice?   (b) How many B. T. U. are required to melt a cake of ice weighing 11 pounds and having a temperature of 17°?
        Ans. (b) 1,667.16 B. T. U.

(1199)   How many B. T. U. are required to raise 6 pounds of superheated steam from 310° to 342°?
        Ans. 92.256 B. T. U.

(1200)   A ball of copper at 305°, weighing 18 pounds, and an iron rod at 278°, weighing 13 pounds, are plunged into a bath of water at 56°.   If the water weighs 32 pounds, what will be its final temperature?   Ans. 77.45°.

(1201)   (a) How many B. T. U. are required to change a pound of water at 212° into steam at atmospheric pressure? (b) How many B. T. U. are required to change 8 pounds of water at 63° into steam at 212°?   Ans. (b) 8,920 B. T. U.

(1202)   How many B. T. U. are required to change 2.2 lb. of ice at 23° into steam at 212°?   Ans. 2,847.98 B. T. U.

(1203)   (a) What is saturated steam?   (b) superheated steam?   (c) In what way does saturated steam differ from a perfect gas?

(1204)   (a) What are the essential features of the horizontal fire-box or locomotive-boiler? (b) of the water-tube boiler?

(1205)   What is the difference between an externally fired flue-boiler and a return tubular boiler?

(1206)   In a fire-box or locomotive-boiler, what method is employed to strengthen the furnace and the external portion of the boiler which surrounds the furnace?

(1207)   Describe four different ways of setting boilers, and how provisions are made for their expansion and contraction.

(1208)   How are boilers generally braced?

(1209)   Do the flues and tubes used in boilers diminish their strength? Give reasons.

(1210)   What is about the level at which the water should stand in a horizontal cylindrical boiler?

(1211)   What are water-cocks? Why should they be mounted upon every boiler?

(1212)   (a) What is air? (b) Does the nitrogen of the air tend to increase or diminish the temperature of combustion in a furnace? Why?

(1213)   Can the combustion of fuels take place without the presence of oxygen?

(1214)   When hydrogen is burned, what does it form with the oxygen of the air?

(1215)   How much heat is generated by the combustion of the carbonic oxide gas, formed from one pound of carbon, to carbonic acid gas?

(1216)   What is the total amount of heat required to convert a pound of water at 32° F. into steam at 400° F.; or, in other words, what is the total heat of evaporation of one pound of saturated steam of 400° F.? Ans. 1,203.4 B. T. U.

(1217)   What is the total amount of heat required to convert a pound of water at 32° F. into steam at a gauge-pressure of 175 pounds per square inch; or, in other words, what is the total heat of evaporation of one pound of saturated steam at an absolute pressure of 189.7 pounds per square inch?

NOTE.—Find temperature first by formula 138.

Ans. 1,198.6 B. T. U.

(1218) If we have 5 cubic feet of saturated steam in a cylinder at 60 pounds pressure above a vacuum, what will be its pressure after it has expanded to 2.5 times its original volume, assuming the expansion to follow Mariotte's law?

Ans. 9.3 pounds per square inch above the atmospheric pressure.

(1219) If 11 pounds of coal are burned per square foot of grate surface per hour in a furnace having a grate area of 13 square feet, how many B. T. U. will be generated in 5 hours, if the combustion of the coal is complete?

Ans. 10,105,095 B. T. U.

(1220) How much air would have to be supplied to promote the complete combustion of the coal in Question 1219, if the furnace is operated under a blast draft?

Ans. 10,010 lb.

(1221) What is the equivalent of the heat of combustion of the fuel in Question 1219, expressed in pounds of water evaporated from 62° F. and at 212° F.?

Ans. 9,059.05 lb. of water.

(1222) The pressure in a boiler is 3,600 pounds per square foot above a vacuum; what is the pressure in the boiler measured in pounds per square inch above the atmospheric pressure?   Ans. 10.3 lb. per sq. in.

(1223) Does saturated steam contain the same amount of heat per unit of weight at all pressures?

(1224) If a vertical boiler were generating steam at a gauge-pressure of 152 pounds per square inch, what would be the temperature of the water in the boiler?

Ans. 371.62° F.

(1225) On placing a thermometer in a jet of steam issuing from a blow-off pipe, we find its temperature to be 232° F.; what is the pressure behind the steam?

Ans. 5.57 lb. per sq. in. gauge-pressure.

(1226) If a coal-mine having a shaft 296 feet deep has an output of 132 tons of coal per hour, how many of the British

Thermal Units of heat supplied to the hoisting-engines with the steam are consumed in raising this coal from the bottom of the shaft ?   Ans. 100,442.15 B. T. U. per hour.

(1227) By the combustion of fuel in the furnace of a boiler, the steam generated during a run of two hours absorbed 277,160 British Thermal Units of heat; if none of the heat had been lost in its transformation into work through the medium of the hoisting-engine to which it was supplied, how many foot-pounds of work per hour would the engine have done ?   Ans. 107,815,240 ft.-lb. per hr.

(1228) A water-tube boiler is built up of a series of 4-inch lap-welded tubes, which are expanded into cast-iron headers through accurately cut holes.   The steam and water drums are 24 and 20 inches in diameter respectively, and are made of single-riveted steel boiler-plate $\frac{5}{16}$ of an inch thick.   The mud-drum is made of cast iron, and is only 10 inches in diameter.   What is the greatest safe boiler-pressure under which the boiler can be operated ?
Ans. 216.25 pounds per square inch above atmospheric pressure.

(1229) A horizontal return tubular boiler has a water-heating surface of 1,620 square feet; what is the approximate horsepower of the boiler ?   Ans. 101¼ H.P.

(1230) A water-tube boiler has a total water-heating area of 3,025 square feet; what is the probable horsepower of the boiler ?   Ans. 275 H. P.

(1231) The sum of the cross-sectional areas of all the tubes of a 348-horsepower fire-box tubular boiler amounts to 12 square feet; what should be the height of a chimney for this boiler to produce the necessary amount of draft ?
Ans. 111 ft.

(1232) Describe two methods of drying steam before it finally leaves the boiler.

(1233) What is the difference between a chimney and a

forced or blast draft? What advantage has the latter over the former?

(1234) What means are usually supplied to facilitate the cleaning of boilers?

(1235) Why are steam-gauges a necessary part of every boiler?

(1236) Why should the water in a boiler be prevented from getting low while the furnace is in full operation?

(1237) Why are internally fired boilers usually bricked in?

(1238) How is the masonry work about a boiler usually strengthened?

(1239) Where should firebrick be used when setting a boiler?

(1240) Describe three different kinds of grates with which you are familiar.

(1241) What is a steam-pipe? a feed-water pipe? a blow-off pipe?

(1242) What are safety-valves? Describe the principle upon which they are operated.

(1243) How far must a 54-pound weight be placed from the fulcrum of a safety-valve that has an area of 6 square inches and is 2 inches from its fulcrum, if the valve is to blow off at 81 pounds per square inch?   Ans. 18 in.

(1244) The shell of a plain cylindrical boiler is 30 inches in diameter and 20 feet long, and is made of single-riveted wrought-iron boiler-plate ⅜ of an inch thick; what is the greatest boiler-pressure under which it can be safely operated?   Ans. 127.8 lb. per sq. in.

(1245) (a) What is meant by the horsepower of a boiler? (b) What is the standard horsepower?

(1246) (a) What is meant by the term *heating-surface?* (b) What portions of an ordinary vertical boiler are heating-surface?

(1247)   If you were placed in charge of a flue-boiler 45 inches in diameter, made of $\frac{4}{16}$-inch double-riveted iron plates, would you consider it safe to carry 110 pounds pressure ?

(1248)   A vertical boiler is rated at 35 horsepower.   (*a*) What is the probable grate-surface ?   (*b*) What is its probable heating-surface ?   (*c*) Under ordinary conditions, how much water per hour would this boiler evaporate, taking the temperature of the feed at 100° and the steam-pressure at 70 pounds ?
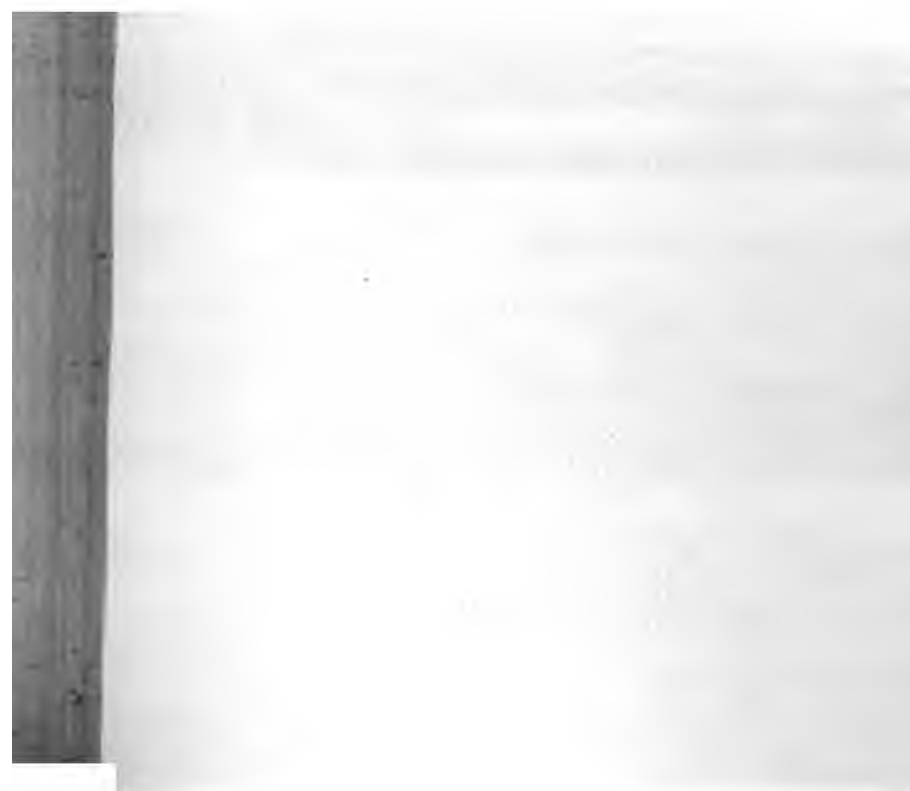
STEAM-ENGINES.

EXAMINATION QUESTIONS.

(1249) Name the stationary parts of a plain slide-valve engine.

(1250) In an indicator-diagram, what is represented by the expansion-curve of steam ?

(1251) Between what points does the plain slide-valve pass the central position of its travel ?

(1252) What is the usual range of cut-off of the plain slide-valve ?

(1253) In Figs. 807, 808, 809, and 810 are given two sets of indicator-diagrams taken from the same engine when running under full load and no load, respectively. Determine from each diagram the steam-pressure in the cylinder of the engine at the point of cut-off. Also the pressure at the point of release and at the point of compression. What is the back-pressure in each case ?

(1254) What was the M. E. P. in the cylinder of the engine at the time the diagrams shown in Figs. 807 and 808 were taken ?   Ans. 43.29 pounds per sq. in.

(1255) What was the M. E. P. in the cylinder of the engine at the time the diagrams shown in Figs. 809 and 810 were taken ?   Ans. 14.96 pounds per sq. in.

(1256) The diagrams Figs. 807 and 808 were taken from an engine with a cylinder 15 inches in diameter, with 24-inch stroke, and making $87\frac{1}{2}$ rev. per minute. Using the M. E. P. found in Question 1254, find the indicated horse-power of the engine.   Ans. I. H. P. = 81 14.

(1257) When working under no load, the engine of Question 1256 gives the diagrams shown in Figs. 809 and 810.

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Scale of Spring 66 lb.

FIG. 807.

Scale of Spring 45 lb.

FIG. 808.

Scale of Spring 45 lb.

FIG. 809.

Scale of Spring 45 lb.
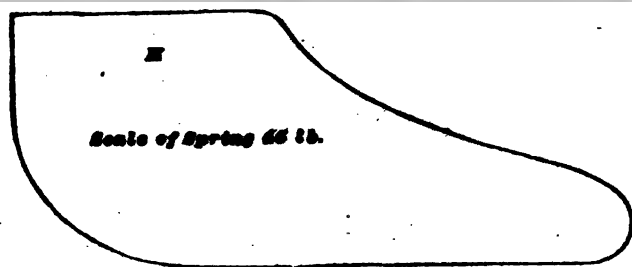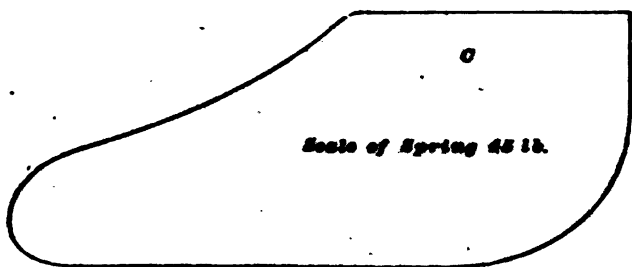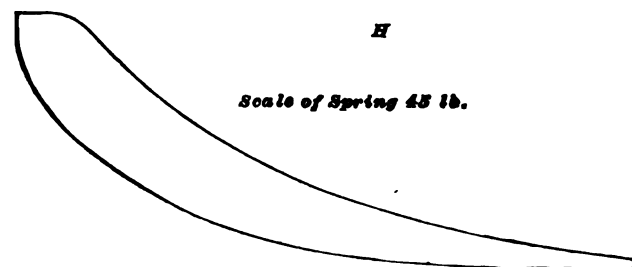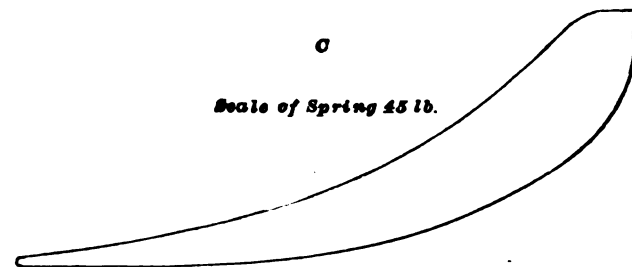
FIG. 810.

The M. E. P. is 14.96 lb.   What is the indicated horsepower in this case ?   Ans. I. H. P. = 28.04.

(1258)   What is the actual horsepower and efficiency of the engine mentioned in Questions 1253 to 1257 ?

Ans. $\begin{cases} \text{Actual horsepower} = 53.1. \\ \text{Efficiency} = 65.4 \text{ per cent.} \end{cases}$

(1259)   What are the forces which, acting on the fly-balls of a pendulum-governor, cause them to move up and down as the speed of the engine varies ?

(1260)   What is a triple-expansion engine ?

(1261)   Name the parts of a plain slide-valve engine to which motion is imparted when the engine is running.

(1262)   Why are engines supplied with fly-wheels ?

(1263)   At the *point of release* of steam from the crank end of the cylinder, what is occurring in the head end of the cylinder ?

(1264)   What is the steam-lap of a plain slide-valve, and why is it given to the valve ?

(1265)   Determine from the indicator-diagrams, Figs. 807, 808, and 809, 810, at what point in the stroke cut-off occurs.

(1266)   Determine approximately the dimensions of a single-cylinder non-condensing engine to furnish 65 actual horsepower.

(1267)   What is the difference between a duplex and a compound engine ?

(1268)   What is a vertical or upright engine ?

(1269)   What is the *stroke* of an engine, and to what is it equal ?

(1270)   What is an eccentric, to what is it equivalent, and what duty does it perform ?

(1271)   What is meant by the period of compression, and when does it occur ?

(1272)   What is the effect of giving inside or exhaust lap to a plain slide-valve ?

(1273)   What is the advantage of the Corliss valve-gear over the plain slide-valve?

(1274)   Find the I. H. P. of an 18-in. × 24-in. engine whose mean effective pressure is 62.4 pounds per square inch, and which makes 175 revolutions per minute.

Ans. 336.825 I. H. P.

(1275)   When an engine has two cranks, why are they placed at right angles to each other on the shaft?

(1276)   What initial steam-pressure is generally used in compound and triple-expansion engines, and how many expansions of the steam are usually effected in each type?

(1277)   What is meant by the *bore* of a cylinder?

(1278)   When is steam called *live steam*, and in what form is energy stored in the live steam?

(1279)   How is the resistance offered by the steam in the cylinder during the period of compression overcome?

(1280)   (*a*) What are the "dead-center" positions of the crank and piston?   (*b*) How many times is the crank on a dead-center during one revolution of the fly-wheel?

(1281)   What is a steam-engine indicator, and how is it attached to the cylinder of a steam-engine?

(1282)   Why do duplex, cross-compound, and triple-expansion engines usually run more smoothly than single-cylinder or tandem-compound engines?

(1283)   What is the *counterbore* of a cylinder, and why is counterbore given to a cylinder?

(1284)   What is meant by *back-pressure?*

(1285)   What is meant by the period of release, and what point marks its end?

(1286)   In what direction should the fly-wheel be rotated when determining the dead-center positions of the crank?

(1287)   How many springs are there in an indicator, and what is the use of each?

(1288)   If a condenser capable of producing a ¼ vacuum had been used in connection with the engine from which

the cards shown in Figs. 807, 808 and 809, 810 were taken, what would have been the effect upon the back-pressure line of the cards?

(1289)   How are hoisting and tail-rope haulage-engines governed?

(1290)   What is meant by *clearance* of a steam-cylinder?

(1291)   How is motion imparted to the slide-valve?

(1292)   During the period of expansion in the crank end if the cylinder, what occurs in the head end of the cylinder?

(1293)   What does the compression-curve show?

(1294)   In what direction should the fly-wheel be rotated rhen setting a plain slide-valve?

(1295)   What is meant by a *40-pound*, a *20-pound*, or a *5-pound* indicator-spring?

(1296)   What is the scale of an indicator-spring?

(1297)   What is the advantage of using the condensed team from a condenser, as boiler feed-water?

(1298)   What outlet is provided in steam-cylinders for the lischarge of water that may accumulate as the result of he condensation of steam?

(1299)   What is the angle between the crank and eccen-ric?

(1300)   If a valve has a slight lead, does the point of .dmission occur at the beginning or end of the stroke?

(1301)   What conditions must be fulfilled in setting a)lain slide-valve?

(1302)   Why is it necessary to employ a reducing motion n connection with an indicator?

(1303)   If a non-condensing engine is working under a)oiler-pressure of 75 pounds per square inch, what is the ipproximate M. E. P. if the engine cuts off at $\frac{3}{10}$ stroke? it $\frac{1}{2}$ stroke?
          Ans. 41.86 and 53.16 lb. per sq. in., respectively.

(1304)   What is the principle that insures the action of :team-engine governors?

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(1305) What is the difference between first and second motion hoisting-engines?

(1306) Why is a piston supplied with split rings?

(1307) What is meant by the *point of cut-off?*

(1308) If a valve has no lead, is the steam-port opened or closed when the crank is on its dead-center?

(1309) What is meant by a valve having *lead?*

(1310) An engine has a piston speed of 350 feet per minute, and makes 175 revolutions per minute; what is the length of the stroke?   Ans. 12 inches.

(1311) Explain the relative duties of a governor and fly-wheel in effecting the regulation of the speed of the engine.

(1312) Explain the action of the compound and of the triple-expansion engine.

(1313) What are stuffing-boxes, and why are they a necessary part of every engine?

(1314) What is meant by the *atmospheric line?*

(1315) How do you determine the length of the stroke and point of cut-off of an engine?

(1316) It is desired to take an indicator-diagram 3 inches in length from an engine of which the length of the stroke is 12 inches and the effective length of the reducing-lever is 96 inches; what is the distance of the point on the lever below the center of the fulcrum at which the cord is to be attached?   Ans. 24 inches.

(1317) An engine has a stroke of 48 inches, and makes 50 revolutions per minute; what is the piston speed?
                    Ans. 400 feet per minute.

(1318) What takes the place of the fly-wheel in hoisting and haulage engines, which have no fly-wheels?

(1319) What is the difference between a tandem and a cross-compound engine? What are the advantages of the tandem type? of the cross-compound type?

(1320) What is the *period of expansion*, and what points mark its beginning and end?

(1321) Between what points are the steam-ports fully open by the valve to the admission of live steam into the cylinder ?

(1322) Find the I. H. P. developed by a 22-in. × 18-in. engine making 200 revolutions per minute. The M. E. P. is 43.4 lb. per sq. in.   Ans. 300 I. H. P.

(1323) What is meant by the term *mean effective pressure ?*

(1324) An engine has a piston speed of 750 feet per minute and a stroke of 60 inches; how many revolutions does the crank make per minute ?   Ans. 75.

(1325) What is the difference between an automatic and a throttling governor ?

(1326) What are the advantages to be gained by compounding ?

AIR AND AIR COMPRESSION.

EXAMINATION QUESTIONS.

(1327)   What do you understand by tension of gases ?

(1328)   A cylinder filled with compressed air supports a column of mercury 4 feet high; (*a*) what is the tension of the air in pounds per square inch ? (*b*) in atmospheres ? Take the weight of a cubic inch of mercury in all cases as .49 pound.

Ans. $\begin{cases} (a) \ 23.52 \text{ lb.} \\ (b) \ 1.6 \text{ atmos.} \end{cases}$

(1329)   By reason of a partial vacuum, a column of water 15 feet in height is supported; what is the tension of the confined air in pounds per square inch ?

Ans. 8.246 lb. per sq. in.

(1330)   What are the advantages to be derived from using compressed air in mining operations ?

(1331)   Why should cold free air be used for compression ?

(1332)   Suppose that air was compressed adiabatically and used immediately, expanding adiabatically back to the original pressure.   Would there be any loss due to adiabatic instead of isothermal compression in this case ?   Why ?

(1333)   (*a*) What is a *wet* air-compressor ?   (*b*) What are its advantages and disadvantages ?

(1334)   Describe the duplex air-compressor, and explain how the arrangement affects the distribution of power between the two compressors.

(1335)   Describe the compound air-compressor, and show wherein lies its advantage over the single-cylinder compressor.

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(1336)   Describe a device for preventing the moisture in the exhaust-air of a compressed-air engine from freezing.

(1337)   (*a*) What is an adiabatic curve ?   (*b*) Which requires the more work, adiabatic compression or isothermal compression, other conditions being the same ?

(1338)   The temperature of the discharged air of an air-compressor, the tension of which is 40 pounds per square inch, is 120° ; when it has cooled down to the temperature of the surrounding air, which is 55°, what is its tension ?

Ans. 35.51 lb.

(1339)   The stroke and diameter of the piston of a blowing-engine (one form of an air-compressor) are each 80 inches.   The valves are so set that they will open for discharge when the tension of the compressed air becomes 9 pounds above the atmosphere.   (*a*) At what point of the stroke will the valves open ?   (*b*) How many cubic feet of air having this tension will be discharged during one stroke of the piston, the temperature being constant throughout ?

Ans. $\begin{cases} (a) & 30.38 \text{ in.} \\ (b) & 144.34 \text{ cu. ft.} \end{cases}$

(1340)   What should be the size of the steam-cylinder of an air-compressor required to furnish sufficient power to drive a 25-horsepower plant ?   Assume the boiler-pressure to be 92 pounds, the cut-off to be $\frac{5}{8}$, the ratio of the stroke to the cylinder diameter to be $1\frac{2}{8}$, the number of strokes 340 per minute, and the loss of power 35%.

Ans. $8\frac{3}{8}$ in. × $11\frac{1}{2}$ in., nearly.

(1341)   If indicator-diagrams are taken at the same time from the steam-cylinder and the air-cylinder of a compressor, which will show the greater indicated horsepower ?   How can you account for the difference between the horsepowers shown by the diagrams ?

(1342)   Describe the electric reheater.

(1343)   The diameter and stroke of the piston of an air compressor are 20 inches and 32 inches respectively.   If the discharge-valve opens when the piston has completed 26

inches of its stroke, (*a*) what is the volume of the contained air ? (*b*) the weight of the air ? The temperature remains at 76° throughout.

Ans. $\begin{cases} (a) \ 1,884.96 \text{ cu. in.} = 1.0908 \text{ cu. ft.} \\ (b) \ .43143 \text{ lb.} \end{cases}$

(1344) In example 1343, what is the tension of the air discharged ?   Ans. 78.4 lb. per sq. in.

(1345) A closed vessel fitted with a piston contains air under a pressure of three atmospheres. If the piston is so moved that the volume is 2½ times its former volume, what is the tension of the gas in pounds per square inch ? The temperature is the same in both cases.

Ans. 17.64 lb. per sq. in.

(1346) A certain quantity of air, under a pressure of 1½ atmospheres and a temperature of 75°, weighs 7.14 pounds; what is its volume ?   Ans. 64.068 cu. ft.

(1347) A certain quantity of air under a pressure of 3½ atmospheres weighs 13 pounds. After expanding under a constant temperature, the weight of the same quantity is only 2 pounds. What is the tension of the air ?

Ans. 7.915 lb. per sq. in.

(1348) The stroke of a piston of an air-compressor is 60 inches. When the piston has traveled 50 inches, what is the tension (the temperature at discharge being 130°) of the enclosed air, assuming that the delivery-valves do not open until this point is reached ? The original temperature is 60°. The diameter of the piston is 48 inches. Obtain the weight of the air, and then calculate the tension.

Ans. 100.096 lb. per. sq. in.

(1349) A pound of air has a temperature of 127° and a tension of 27 pounds per square inch. What is its volume ?

Ans. 8.042 cu. ft.

(1350) The weight of a certain body of air having a tension of 4,000 pounds per square foot and a temperature of 100° is .5 pound. What is its volume ?

Ans. 3.728 cu. ft.

(1351) Four cubic feet of air are heated under a constant pressure from 40° to 115°. What is the resulting volume?

Ans. 4.6012 cu. ft.

(1352) State the advantages of cooling air during compression; of reheating it.

(1353) What are the absolute temperatures corresponding to 32°, 212°, 62°, 0°, and —40°?

(1354) Three and one-half pounds of air under a pressure of 10 atmospheres occupy a volume of 4 cubic feet; what is the temperature?   Ans. —5.583°.

(1355) State some of the disadvantages of the duplex type of air-compressor.

(1356) 11.798 cubic feet of air are under a pressure of 130 pounds per square inch. If the pressure is lessened until the volume is 75 cubic feet, what is the resulting tension?   Ans. 20.45 lb. per sq. in.

(1357) What is the temperature of 14 cubic feet of air having a tension of 18 pounds per square inch and weighing 1.2 pounds?   Ans. 107.77°.

(1358) Twenty-one cubic feet of air are heated from 60° to 420°; what is the new volume?   Ans. 35.57 cu. ft.

(1359) If 12 cubic feet of air have a temperature of 90° and a tension of 6 atmospheres gauge, what is the weight of 1 cubic foot?   Ans. .50586 lb.

(1360) A vessel containing 3 cubic feet of air, weighing .5 pound under a pressure of one atmosphere, has compressed into it enough more of the air to make it weigh 1 pound and 6 ounces; the temperature remaining the same, what is the new tension of the air in pounds per square inch?

Ans. 40.425 lb. per sq. in.

(1361) If 4,516 cubic inches of gas having a temperature of 260° are cooled down to a temperature of 80°, the pressure remaining the same, what is the new volume?

Ans. 1.96 cu. ft

(1362) If 55 cubic feet of air under a pressure of 1¼ atmospheres have a temperature of 88°, what is the weight ?
Ans. 4.986 lb.

(1363) Two vessels, the volumes of which are each 7½ cubic feet, are filled with air; the temperature is the same in both, but the pressure in one is two atmospheres, and in the other 40 pounds per square inch.   If all of the air in one vessel is compressed into the other, what is the pressure of the mixture after it has cooled down to the original temperature ?   Ans. 69.4 lb. per sq. in.

(1364) If you are told that the vacuum-gauge of a condenser shows 23 inches vacuum, what do you understand by it ?   What is the pressure in the condenser ?

(1365) What is a pressure of one atmosphere equivalent to in pounds per square foot ?   Ans. 2,116.8 lb. per sq. ft.

(1366) If the weight of 3 cubic feet of air at a certain temperature and under a pressure of 30 pounds per square inch is .27 pound, what is the weight of one cubic foot under a pressure of 65 pounds per square inch at the same temperature ?   Ans. 0.195 lb.

(1367) In example 1366, what is the temperature of the air ?   Ans. 440.64°.

(1368) Two gases, oxygen and nitrogen, are mixed together in a vessel containing 20 cubic feet.   The volume and tension of the oxygen are 12 cubic feet and one atmosphere, respectively, and of the nitrogen 8 cubic feet and three atmospheres.   The temperature of the two gases and of the mixture remaining the same throughout, what is the tension of the mixture ?   Ans. 26.46 lb. per sq. in.

(1369) In example 1368, suppose that the volume of the mixture is not known, and that the pressure is required to be 24 pounds per square inch; what is the volume of the mixture ?.   Ans. 22.05 cu. ft.

(1370) What is a vacuum ?   Illustrate it.

(1371) An air-pump produces a vacuum of $\frac{1}{40}$ of an inch

of mercury; what is the equivalent pressure upon a square foot ?   Ans. 1.764 lb.

(1372) What is a *partial vacuum?* If enough air is admitted to the vacuum-chamber to cause the column of mercury to be 4½ inches shorter than the barometer column, how many inches of vacuum will the gauge show ?

                                          Ans. 25½ in.

(1373) A vacuum of 27 inches will support a column of water of what height?   Ans. 30.6 ft.

(1374) What is the purpose of a pressure-regulator? Describe its action.

(1375) What is the office of the receiver ? What should be the volume of a receiver which supplies air to 8 rock-drills ?

(1376) What is meant by the *efficiency* of an air-compressor ? State fully the losses which may occur when compressed air is used. What means should be adopted to reduce these losses as far as possible ?

HYDROMECHANICS AND PUMPING.

EXAMINATION QUESTIONS.

(1377) A weir whose top is 3 feet 6 inches below the surface of the water is 2 feet deep and 30 inches broad; (*a*) what is the actual mean velocity ? (*b*) What is the discharge in cubic feet per second ? (*c*) in gallons per hour ?

Ans. $\begin{cases} (a) \ \ 10.44 \text{ ft. per sec.} \\ (b) \ \ 52.21 \text{ cu. ft. per sec} \\ (c) \ \ 1,405,910.9 \text{ gal. per hour.} \end{cases}$

(1378) A pipe 12,000 feet long and $7\frac{1}{2}$ inches in diameter discharges water under a head of 76 feet; what is the discharge in gallons per minute ?   Ans. 447.7 gal.

(1379) In the last example, (*a*) what is the velocity of discharge in feet per minute ? (*b*) What is the discharge in cu. ft. per second ?   Ans. $\begin{cases} (a) \ \ 195.08 \text{ ft. per min.} \\ (b) \ \ 1 \text{ cu. ft., nearly.} \end{cases}$

(1380) An 8-inch pipe has a hole in it $\frac{1}{2}$ of an inch in diameter; what would be the theoretical velocity of efflux if the surface of the water were 10 feet above the center of the hole ?   Ans. 25.36 ft. per sec.

(1381) What must be the necessary head in order that a $6\frac{1}{2}$-inch pipe, 1,500 feet long, shall discharge 42,000 gallons of water per hour ?   Ans. 42.48 ft.

(1382) A vertical cylinder having a diameter of 20 inches and a length inside of 36 inches is filled with water. A pipe having a diameter of $\frac{3}{8}$ of an inch is screwed into the upper head, and fitted with a piston weighing 10 ounces, on which is laid a weight of 25 pounds. If the end of the pipe is 10

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feet above the level of the water in the cylinder, (*a*) what is the pressure per sq. in. on the bottom of the cylinder? (*b*) on the top? (*c*) What equivalent weight laid on the lower cylinder-head would replace the pressure it sustains?

Ans. $\begin{cases} (a) & 237.75 \text{ lb.} \\ (b) & 236.45 \text{ lb.} \\ (c) & 74,692.17 \text{ lb.} \end{cases}$

(1383)   If, in Question 1382, a hole one inch in diameter is drilled through the cylinder-wall in the middle of its length, and is covered by a flat plate in such a manner that the water can not leak out, what is the pressure against the plate?   Ans. 186.22 lb.

(1384)   What is the mean velocity of efflux from a straight pipe 4 inches in diameter and 4,000 feet long, under a head of 120 feet?   Ans. 5.38 ft. per sec.

(1385)   If the length of the pipe in Question 1384 had been 2,000 feet, what would the velocity of discharge have been in feet per second?   **Ans. 7.79 ft. per sec.**

(1386)   A 10-inch pipe 5,280 feet long is required to deliver water with a velocity of 8 feet per second; (*a*) what is the necessary head? (*b*) What is the discharge in gallons per hour?   Ans. $\begin{cases} (a) & 130.73 \text{ ft.} \\ (b) & 117,504 \text{ gal. per hour.} \end{cases}$

(1387)   What is the actual velocity of discharge from a small square-edged orifice in the side of a vessel, if the water at the center of the orifice has a pressure of 30 pounds per square inch?   Ans. 65.34 ft. per sec.

(1388)   The upper base of a cylinder submerged in water is 40 feet below the surface. The diameter of the cylinder is 20 inches, the altitude 36 inches, and the bases are parallel. If the bases are horizontal, (*a*) what is the upward pressure of the water on the cylinder? (*b*) the downward pressure on the top of the cylinder?   Ans. $\begin{cases} (a) & 5,863.26 \text{ lb.} \\ (b) & 5,454.19 \text{ lb.} \end{cases}$

(1389)   A jet of water issues with a velocity of 33 feet per second; what would be the theoretical head necessary to give it this velocity?   **Ans. 16.931 ft.**

390) A weir having a depth of 15 inches and a breadth
inches has its top on a level with the upper surface of the
r; (*a*) how many gallons will it discharge per hour? (*b*)
t is the actual mean velocity in feet per second?

Ans. { (*a*) 216,551 gal., nearly.
{ (*b*) 3.676 ft. per sec.

391) A 3-inch pipe, 6,000 feet long, is required to de-
water at a velocity of 12 feet per second; what head is
ssary? Ans. 1,040.37 ft.

392) A 5-inch pipe discharges water with a velocity of
eet per second; how many gallons will it discharge in
lay? Ans. 634,478 gal.

393) A 5½-inch pipe discharges 38,000 gallons of water
our; what is the mean velocity in feet per second?
Ans. 8.5526 ft. per sec.

394) A weir whose top is on a level with the upper sur-
of the water is 27 inches broad and 36 inches deep; (*a*)
is the actual discharge in cubic feet per second? (*b*)
t is the theoretical discharge? Ans. { (*a*) 38.44 cu. ft.
{ (*b*) 62.5 cu. ft.

395) If the surface of the water in a 6-inch pipe is 45 feet
e the discharge-orifice, which is 1½ inches in diameter, (*a*)
will be the theoretical velocity of efflux? (*b*) If the
r surface of the water sustains an additional pressure of
unds per square inch, what will be the velocity of efflux?

Ans. { (*a*) 53.8 ft. per sec.
{ (*b*) 66.153 ft. per sec.

396) What is the discharge in gallons per second from
ich pipe, if the mean velocity is 7.5 feet per second?
Ans. 11.016 gal.

97) What is the actual velocity of discharge through
rt tube whose length is twice the diameter of the orifice,
pressure of the water at the point of discharge is 41
ds per square inch? Ans. 76.39 ft. per sec.

(1398) A 4-inch pipe discharges 12,000 gallons per hour; what is the mean velocity of discharge in feet per second?
Ans. 5.106 ft. per sec.

(1399) The cylinder of a hydraulic press is 10 inches in diameter. The plunger is forced outwards by means of a small pump which supplies the press-cylinder with water, its piston being ¼ inch in diameter and stroke 1½ inches. If a force of 100 pounds is applied to the pump-piston, (a) how great a force can it exert on the plunger? (b) How far does the plunger advance for one stroke of the piston?

Ans. $\begin{cases} (a) \ 40,000 \text{ lb.} \\ (b) \ .00375 \text{ in.} \end{cases}$

(1400) How many gallons per minute will a weir 14 inches by 20 inches discharge, if the top of the weir is 9 feet below the upper surface of the liquid, (a) when the long side is vertical? (b) when the short side is vertical?

Ans. $\begin{cases} (a) \ 13,491.22 \text{ gal.} \\ (b) \ 13,322.47 \text{ gal.} \end{cases}$

(1401) What is the mean velocity for both cases of Question 1400?

Ans. $\begin{cases} (a) \ \textbf{15.46 ft. per sec.} \\ (b) \ \textbf{15.264 ft. per sec.} \end{cases}$

(1402) What is the theoretical mean velocity of discharge through a weir whose depth is 3 feet and whose top is level with the upper surface of the water? Ans. 9.26 ft. per sec.

(1403) The surface of the water contained in a vessel is 19 feet above the ground; (a) what is the range of the water issuing from an orifice 4 feet 9 inches from the top? (b) How far below the surface is the other point of equal range? (c) What is the greatest range?

Ans. $\begin{cases} (a) \ \textbf{16.454 ft.} \\ (c) \ \textbf{19 ft.} \end{cases}$

(1404) A 5-inch pipe 1,300 feet long discharges water under a head of 25 feet; what is the number of gallons discharged per hour?   Ans. 17,368.95 gal.

(1405) What values of f would you use for $v_m = 2.37$, 3.19, 5.8, 7.4, 9.83, and 11.5, respectively?

(1406) What would be the total pressure on a cube, one

edge of which measures 10½ inches, if sunk 3½ miles below sea-level ?

(1407) The diameter of the bottom of a pail is 8 inches, and the height of the contained water is 12 inches; (*a*) what is the total pressure on the bottom of the pail ? (*b*) What is the pressure per square inch ?

Ans. { (*a*) 21.82 lb.
{ (*b*) .434 lb. per sq. in.

(1408) What must be the diameter of a pump-plunger to throw 8,000 gallons per hour, the length of the stroke being 10 feet, and the number of strokes per minute 7 ? Ans. 7⅝ in.

(1409) What are the advantages of the pulsometer ?

(1410) If a suction-pump lifts water 25.5 feet near the sea-level, where the height of the mercury column is 30 inches, how high will the same pump lift water on the top of a mountain, where the mercury stands at 22 inches ?
Ans. 18.7 ft.

(1411) A dam is 40 feet long and 12 feet high; what is the total pressure on the dam ? Assume that a cubic foot of water weighs 62½ pounds.   Ans. 180,000 lb.

(1412) Calculate the diameters of the plunger, of the suction-pipe, and of the delivery-pipe for a double-acting pump throwing 750 gallons per minute. Assume 100 feet per minute as piston speed.   (Plunger, 15 in.
Ans. { Delivery, 7 in.
(Suction, 10 in.

(1413) The total length of a siphon is 840 feet, the head is 40 feet, and the diameter 6 inches; what is the discharge in gallons per hour ?   Ans. 44,553.6 gal. per hr.

(1414) The lever of a hydraulic press is 7½ feet long, the piston-rod being 1 foot from the fulcrum. The area of the tube is ½ a square inch; that of the cylinder 80 square inches. What weight may be raised by a force of 80 pounds applied at the end of the lever ?   Ans. 96,000 lb.

(1415) A duplex electric sinking-pump lifts 200 gallons

per minute to a height of 250 feet. The piston speed is 150 feet per min.

(a) What should be the diameter of the plunger?
(b) What should be the diameter of the suction-pipe?
(c) What should be the diameter of the delivery-pipe?
(d) What should be the H. P. of the motor?

$$\text{Ans.} \begin{cases} (a) \ 4\tfrac{1}{2} \text{ in.} \\ (b) \ 5 \text{ in.} \\ (c) \ 3\tfrac{1}{2} \text{ in.} \\ (d) \ 19 \text{ H. P.} \end{cases}$$

(1416) A hydrant is 210 feet below the surface of the water-supply; (a) what is the pressure of the water issuing from the hydrant, and (b) what is the theoretical velocity?

$$\text{Ans.} \begin{cases} (a) \ 91.14 \text{ lb. per sq. in.} \\ (b) \ 116.22 \text{ ft per sec.} \end{cases}$$

· (1417) Calculate the size of the steam and water cylinders of a duplex direct-acting steam-pump to lift 27,000 gallons per hour to a height of 240 feet. Assume the steam-pressure as 85 pounds per square inch and piston speed as 90 feet per minute.

$$\text{Ans.} \begin{cases} \text{Diameter of steam-cylinder, } 10\tfrac{5}{8} \text{ in.} \\ \text{Diameter of plunger, } 8\tfrac{3}{4} \text{ in.} \end{cases}$$

(1418) A water-works stand-pipe is filled with water to the height of 70 feet; (a) what is the lateral pressure per square inch at the lowest point of the stand-pipe? (b) at a distance of 30 feet from the top of the water?

$$\text{Ans.} \begin{cases} (a) \ 30.38 \text{ lb. per sq. in.} \\ (b) \ 13.02 \text{ lb. per sq. in.} \end{cases}$$

(1419) Why are air-chambers used on pumps?

(1420) The diameters of the steam-cylinders of a duplex direct-acting pump are 22 inches. The diameters of the water-cylinders are 14 inches. The steam-pressure is 45 pounds and the piston speed 100 feet per minute; (a) how many gallons per hour will this pump raise, and (b) to what height?

$$\text{Ans.} \begin{cases} (a) \ 76,769.28 \text{ gal. per hr.} \\ (b) \ 213.22 \text{ ft.} \end{cases}$$

(1421)   The plunger of a Cornish pump is 307 feet below
the mouth of the discharge-pipe; what is the pressure per
square inch on the plunger-cylinder when discharging water
to the surface ?   Ans. 133.238 lb. per sq. in.

(1422)   Why must the pit-work of surface engines be
balanced ?   Explain fully.

(1423)   Water flows through a 2½-inch pipe, 2,000 feet
long, with a velocity of 3.3 feet per second; what is the
head ?   Ans. 39.12 ft.

(1424)   (a) What horsepower is required to raise 80,000
gallons per hour to a height of 420 feet ?   (b) What should
be the horsepower of the pumping-engine to accomplish
this ?   Ans. $\begin{cases} (a) \ 141.87 \text{ H. P.} \\ (b) \ 212.8 \text{ H. P.} \end{cases}$

(1425)   A pump lifts 30,000 gallons of water per hour to
height of 290 feet; 600 pounds of coal per hour are burned;
what is the duty ?   Ans. 12,114,750 ft.-lb.

(1426)   A 6-inch pipe, 6,500 feet in length, has a head of
20 feet; what is the mean velocity of efflux ?
                                     Ans. 7.17 ft. per sec.

(1427)   What head of water corresponds (a) to a pressure
45 pounds per square inch? (b) to 86 pounds per square
inch ? (c) to 108 pounds per square inch ?
                    Ans. $\begin{cases} (a) \ 103.68 \text{ ft.} \\ (b) \ 198.144 \text{ ft.} \\ (c) \ 248.832 \text{ ft.} \end{cases}$

(1428)   The diameter of the water-cylinder of a direct-
acting pump is 15 inches, and the height of lift is 310 feet;
(a) what is the diameter of the steam-cylinder ?   (b) What
will be the delivery at a piston speed of 100 feet per minute ?
Take the steam-pressure as 50 pounds per square inch.
                    Ans. $\begin{cases} (a) \ 27 \text{ in.} \\ (b) \ 734.4 \text{ gal. per min.} \end{cases}$

(1429)   With what velocity, theoretically, will water flow
from an orifice 13.7 feet below the surface ?
                                     Ans. 29.685 ft. per sec.

(1430)   In Fig. 736 the weight on piston a is 22 pounds; the area of a is 5 square inches, and the area of b is 73 square inches; what must be the weight on b to just balance the weight on a ?   Ans. 321.2 lb.

(1431)   Why can water be sucked up through a straw ?

(1432)   The diameter of the water-cylinder of a single direct-acting pump is 11 inches.   The steam-pressure is 50 pounds per square inch, and the height of the lift is 300 feet; (a) find the discharge per hour.   (b) Find the diameter of the steam-piston.   (c) What is the horsepower of the pump ?   Assume the piston speed as 100 feet per minute.

Ans. $\begin{cases} (a)\ 23{,}696.64 \text{ gal. per hr.} \\ (b)\ 19\frac{1}{2} \text{ in.} \\ (c)\ 45.024 \text{ H. P.} \end{cases}$

(1433)   If a piece of glass be laid upon a flat surface which has been moistened, it will require considerable exertion to separate them.   Why ?

(1434)   The total length of a siphon is 88 feet; the head is 15 feet, and the diameter $3\frac{1}{4}$ inches; what is the discharge in gallons per minute ?   Ans. 342.66 gal. per min.

(1435)   What should be the size and proportions of a direct-acting steam-pump to deliver 18,000 gallons per hour against a head of 225 feet ?   Assume the average steam-pressure to be 50 pounds per square inch.   Add one-half to the indicated horsepower for friction, etc., and take the piston speed as 110 feet per minute.

Ans. $\begin{cases} \text{Diameter of steam-cylinder, 14 in.} \\ \text{Diameter of water-cylinder, } 9\frac{1}{4} \text{ in.} \\ \text{Stroke, 12 in.} \\ \text{Diameter of suction-pipe, 6 in.} \\ \text{Diameter of discharge-pipe, } 4\frac{1}{4} \text{ in.} \end{cases}$

(1436)   The area of the cross-section of an orifice in a thin plate is 11.2 square inches.   There being a constant head of 15 feet 9 inches, (a) what is the theoretical discharge

in cubic feet per minute ?   (*b*) What is the actual discharge
in cubic feet per minute ?

Ans. $\begin{cases} (a) \ 148.54 \text{ cu. ft. per min.} \\ (b) \ 91.344 \text{ cu. ft. per min.} \end{cases}$

(1437)   Why must a siphon be filled with water, or have
the air exhausted from it, before it will work ?

(1438)   The diameter of the plunger of a pump is 19
inches; the length of the stroke is 9 feet, and the number
of strokes per minute 5; (*a*) what is the discharge in gal-
lons per minute ? (*b*) per hour ?   Calculate the discharge by
formula **191.**   Ans. $\begin{cases} (a) \ 530.24 \text{ gal. per min.} \\ (b) \ 31,814.4 \text{ gal. per hr.} \end{cases}$

(1439)   A pumping-engine lifts 80,000 gallons per hour
340 feet, with a coal consumption of 400 pounds; what is the
duty ?   Ans. 56,814,000 ft.-lb.

(1440)   Find the heads of water corresponding to the fol-
lowing pressures: (*a*) 80 pounds per square inch; (*b*) 30.5
pounds per square inch; (*c*) 108 pounds per square inch;
(*d*) 215 pounds per square inch.

Ans. $\begin{cases} (a) \ 184.32 \text{ ft.} \\ (b) \ 70.272 \text{ ft.} \\ (c) \ 248.832 \text{ ft.} \\ (d) \ 495.36 \text{ ft.} \end{cases}$

(1441)   The piston speed of a duplex steam-pump is 100
feet per minute; the diameter of the plunger is 14 inches;
what is the delivery in gallons per hour ?

Ans. 76,769.28 gal. per hr.

(1442)   A 4-inch pipe 5,000 feet long is required to
deliver water with a velocity of 8 feet per second; what
must be the head ?   Ans. 307.46 ft.

(1443)   A cylindrical vessel, 3 feet in diameter and 12 feet
long, is placed upon one end, so that its axis is vertical.
Suppose that it is kept filled with water which flows through
a hole in the bottom; what will be the velocity of efflux if
the hole is 11 inches square ?   Ans. 27.979 ft. per sec.

(1444)   A compound condensing pumping-engine delivers

4,000,000 gallons of water in 10 hours, against a head of 125 feet. The number of pounds of coal burned in 10 hours is 7,460; what is the duty?   Ans. 55,998,660 ft.-lb.

(1445) A Cornish pumping-engine has a stroke of 10 feet. The pit-work weighs 20 tons, the water-column 12 tons, and the frictional resistances are 3 tons; what must be the weight of the counterbalance in order that the greatest speed of the pit-work may be about 200 feet per minute?
Ans. 2.6 tons.

(1446) How is the expansion of steam obtained in a compound pumping-engine?

(1447) State some of the advantages of using an electric sinking-pump.

(1448) What should be the diameters of (a) the suction and (b) delivery pipes of a pump which discharges 70,000 gallons of water per hour?   Ans. $\begin{cases} (a) \ 12 \text{ in.} \\ (b) \ 8\frac{1}{2} \text{ in.} \end{cases}$

(1449) What must be the horsepower of a pump to deliver 100,000 gallons of water per hour against a head of 480 feet?   Ans. 304 H. P.

(1450) How many gallons per hour will a 7-inch pipe deliver, if the mean velocity of the water at the point of efflux is 7.21 feet per second?   Ans. 51,891.24 gal. per hr.

(1451) What special advantages does the Cameron sinking-pump possess over other steam sinking-pumps?

(1452) In what cases can a hydraulic pump be used to a great advantage?

(1453) State what is meant by a Cornish pumping-engine; a Bull engine; a sinking-pump; a hydraulic engine; a siphon.

(1454) What is the usual practice in regard to the velocity of the water in the suction-pipe? in the delivery-pipe? What is the usual limit of piston speed in pumps?

(1455) How many gallons of water will a pump deliver

er hour, if the diameter of the pump-cylinder is 15 inches
nd the piston speed is 95 feet per minute ?

<div style="text-align:right">Ans. 41,860.8 gal. per hr.</div>

(1456)   State what is meant by a compound, a duplex, a
ngle compound, an outside packed triplex, and an outside
icked compound condensing duplex pump.

(1457)   A jet of water issues from an orifice under a head
' 69.12 feet; what is the actual velocity in feet per second ?

<div style="text-align:right">Ans. 65.34 ft. per sec.</div>

(1458)   A squirt-gun has a hole in it $\frac{3}{32}$ of an inch in
ameter.   It is held vertically upwards, and a pressure of
) pounds is applied to the piston, which is $\frac{1}{8}$ of an inch in
ameter.   Neglecting all resistances, (*a*) how high will the
ater rise ?   (*b*) If held horizontally 10 feet from the ground,
hat will be its range ?

<div style="text-align:right">Ans. $\begin{cases} (a) & 191.6 \text{ ft.} \\ (b) & 87.54 \text{ ft.} \end{cases}$</div>

MINE HAULAGE.

EXAMINATION QUESTIONS.

(1459) How many stages of underground haulage are there, and which should be the shorter ?

(1460) How many classes of wire-rope haulage are there ? Name them.

(1461) Is it always economy to use a self-acting plane for haulage ? Give reasons.

(1462) In what does the mechanism used on self-acting planes with heavy pitches differ from that used on self-acting planes with light pitches ?

(1463) Which is preferable on a self-acting plane with a light pitch, a two-rope system or an endless-rope system ?

(1464) What is essential when the length of a gravity-plane of light pitch run with a pair of ropes is increased ?

(1465) Under what conditions are gravity-planes of great value in mine haulage ?

(1466) Describe that type of self-acting incline known as a jig.

(1467) On an ordinary self-acting incline run by two ropes, why must one rope run off the bottom of the drum while the other runs off the top ?

(1468) At what point on a self-acting plane run by two ropes do the trains begin to acquire a higher velocity, and why ?

(1469) What mechanisms are sometimes used to replace a drum on a self-acting incline ?

(1470) Under what conditions is it necessary to make the opening for the self-acting plane as narrow as possible ?

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(1471) Why is timbering on a self-acting incline a source of danger and expense?

(1472) What provision should be made on a self-acting incline to prevent a train from running away, in case the rope should break?

(1473) What advantage has a grip-wheel over a double-rope reel?

(1474) How does a grip-wheel hold the rope?

(1475) What are the advantages of geared drums?

(1476) What are deflecting sheaves, and what are their uses?

(1477) What are the disadvantages of a fleet-wheel?

(1478) On a drum of given size, one rope has two turns around the drum and another has four turns; how much more hold or grip has the latter than the former?

(1479) If two wheels are connected by rope belts, and there are two half coils on each wheel, how many coils would that be equal to if but one wheel were used?

(1480) In case you had a self-acting incline of uniform grade that did not work, by what means could that plane be made self-acting; that is, how and where would you change the grade to make it self-acting?

(1481) What is the cause that retards the extension and rapid action of self-acting inclines at small pitches?

(1482) How many varieties of brakes for self-acting drums are there? Describe each.

(1483) What three important points regarding inclination must be considered in the operation of gravity-planes?

(1484) What is the operative force of self-acting inclines?

(1485) If a body weighing 500 pounds moves down an incline through a distance of 200 feet, and falls a vertical height of 20 feet, what weight rising vertically will balance the body moving down the incline?

(1486) How is the pull affected when the rope is not parallel to the incline?

(1487) What two considerations are most prominent in the operation of gravity-planes ?

(1488) If a rope 1,800 feet long weighs 3,500 pounds, and the incline of the plane on which it is used is equal to a grade of 12 per cent., what force is required to move the rope ?   Ans. 507.5 lb.

(1489) How do you find the friction due to an empty car descending a gravity-plane, and a loaded car ascending ?

(1490) A gravity-plane has a grade of 10 per cent. It is 2,500 feet in length, and the rope attached to the empty cars at the foot of the incline weighs 4,200 pounds. A loaded car weighs 4,000 pounds, and an empty one 1,800 pounds. What is the number of cars that must be run in a train to overcome the resistance of the rope at the start of the run ?   Ans. 7 cars.

(1491) On what kind of inclines can the jig system be successfully used ?

(1492) Prove that the jig system can not be used on an incline with 25 per cent. grade, length of road 250 feet, weight of rope per foot of length 1.5 pounds, weight of a full car 4,000 pounds, weight of an empty car 1,800 pounds, and the weight of the jig 2,900 pounds.

(1493) How can the grade be changed in the jig incline mentioned in Question 1492 so as to make it operate ?

(1494) What is the best plan to determine the size of the rope for use on a self-acting incline ?

(1495) What is the tension in a rope when the loaded cars leave the top of the incline under the following conditions: A loaded car weighs 4,000 pounds and an empty car 1,800 pounds. There are two cars in a train, and the grade is one of 25 per cent.; the length of the incline is 1,500 feet, and the weight of the rope per foot of length is .8 pound ?   Ans. 1,320 lb.

(1496) What track arrangements should there be at the head of gravity-planes on which the descending loaded cars raise the empty ones ?

(1497) What precautions should be taken to prevent cars from descending the plane before they are properly attached to the rope?

(1498) What is a safety-lock, and for what is it used?

(1499) Under what conditions are engine-planes adopted for haulage?

(1500) What are the three general classes of engine-planes?

(1501) In what cases are engine-planes superior to other systems of haulage?

(1502) Where, besides at the head of the engine-plane, are the haulage-engines sometimes located?

(1503) What is a *barney*?

(1504) How are sheaves set to carry the rope around a curve?

(1505) What is a drag-bar, and what is its use?

(1506) Under what conditions is the length of an engine-plane haulage such as to make it uneconomical?

(1507) What is the maximum number of cars that should be run in a train, and what is the minimum grade for an engine-plane to attain an average speed of 10 miles per hour, when empty cars are running back?

(1508) If on an engine-plane 18 loaded cars weighing 4,000 pounds each are hoisted with a haulage-rope 5,000 feet long, and weighing .88 pound per foot, what is the tension in the rope at the moment the engine hauls away from the bottom of the incline, the grade being 5 per cent.?

Ans. 5,730 lb.

(1509) If the train in Question 1508 has a velocity of 10 miles per hour, what is the horsepower required to do the work?   Ans. 152.8 H. P.

(1510) Why, in calculating tension on the rope, or the horsepower required, should the full weight of the rope be taken, instead of the average weight?

(1511) Which is more economical: to increase the speed of the haulage so as to make double the number of trips

with half the number of cars and with a lighter rope, or to run at half the speed with double the load and a heavy rope?

(1512) In case the hoisting-shaft is situated in a shallow · basin of such character that the loaded trains will run by gravity to the shaft, and with sufficient fall to haul the empty trains into the different stations in the workings, how can engine-plane haulage be modified so as to form a cheap and efficient system of haulage?

(1513) What horsepower is required to haul 25 empty cars along an incline 3,000 feet long, on a grade of 5 per cent., each empty car weighing 1,500 pounds, the weight of the rope per foot of length being .88 pound, and the maximum velocity of the train being 12 miles per hour?

Ans. 94.2 H. P.

(1514) On what classes of roads can the tail-rope system of haulage be adopted with success?

(1515) In what general feature does the tail-rope system of haulage differ from other systems?

(1516) Which is called the main rope and which the tail-rope in the tail-rope system of haulage?

(1517) What are the general features of tail-rope haulage?

(1518) In running a tail-rope haulage system, what are the duties of the engineer to insure the successful operation of the system?

(1519) · Under what conditions are geared engines more economical for tail-rope haulage than direct-motion engines, and under what conditions is the direct-motion engine preferable?

(1520) How can a tail-rope haulage be made to haul from two or more different districts in the mine?

(1521) Are the main and tail ropes of a tail-rope haulage always of the same weight per linear foot? If not, why not?

(1522) The greatest length of main and tail rope haulage in a certain mine is 7,000 feet, and the tracks are

perfectly level; the weight per foot of the main rope is .7 pound, the weight per foot of the tail-rope is .6 pound, the full cars weigh 4,500 pounds, the empty cars 1,500 pounds, and the trains consist of 20 cars. (*a*) What is the tension on the main and tail ropes? (*b*) If the average speed of trains is 10 miles per hour, what is the horsepower of the hauling engines due to the maximum tension in the ropes?

Ans. (*b*) 66.1 H. P.

(1523) In a short portion of a mine with tail-rope haulage, the main rope must haul a train of 20 loaded cars up a grade of 3 per cent.; what is the maximum tension in the main rope when a full car weighs 4,500 pounds, the main rope weighs 1.2 pounds per foot, the tail-rope .88 pound per foot, and the length of the track is 6,000 feet?

Ans. 5,262 lb.

(1524) If in a level haulage 6,000 feet long there is a slight up grade 60 feet in length, and the power required for the level road is 90 horsepower, what is the increased power exerted in hauling the load up the short grade?

Ans. 90.9 H. P.

(1525) All the roads of a tail-rope haulage leading to the shaft have a mean fall of 4 per cent. The greatest length of run is 4,000 feet, the mean velocity is 11 miles per hour, the haulage-rope weighs .88 pound per foot, the trains consist of 20 cars, each loaded car weighing 5,000 pounds, and each empty car 2,000 pounds. (*a*) What are the tensions in the main and (*b*) the tail rope, respectively? (*c*) What is the horsepower of the haulage-engine?

Ans. $\begin{cases} (a) & - 1,324 \text{ lb.} \\ (b) & 2,776 \text{ lb.} \\ (c) & 81.4 \text{ H. P.} \end{cases}$

(1526) If in Question 1525 the main rope and tail-rope weigh .6 pound and 3.65 pounds per foot, respectively, (*a*) what will be the tension in each rope, and (*b*) what will be the required horsepower of the haulage-engine?

Ans. (*b*) 74.4 H. P.

(1527)   What should be the maximum velocity of trains in main and tail rope haulage, and why?

(1528)   What factors must be determined before assuming the number of cars that should be attached in a train in tail-rope haulage?

(1529)   How are these factors found?

(1530)   How many trains can be run out by a main and tail rope haulage in one day of 10 hours, the speed of the rope· being 12 miles per hour, and the length of the five districts being as follows: *a*, 6,000 feet; *b*, 4,800 feet; *c*, 2,500 feet; *d*, 7,000 feet; *e*, 3,000 feet?

(1531)   If the output of a mine is 2,500 tons of coal per day, and the number of trains to haul out this quantity is 46, and if each car carries 2.5 tons, how many cars must be used to do the work?   Ans. 22 cars.

(1532)   Find (*a*) the number of trains that can be run out per day, (*b*) the number of cars in a train, and (*c*) the horse-power of the engine of a main and tail rope haulage, for an output of 2,500 tons in 10 hours.   The coal has to be hauled out of 4 districts, *a*, *b*, *c*, and *d*, whose lengths are, respectively, 4,250 ft., 3,012 ft., 756 ft., and 514 ft.   The average up grade of the roads is 3 per cent. towards the shaft.   The cars carry 2.5 tons each, an empty car weighs 2,000 pounds, the weight of the rope is 1.5 lb. per foot, and the speed of the trains is 11 miles an hour.   Allow $\frac{1}{3}$ of the time for delays in haulage.

Ans. (*a*) 91 trains.   (*b*) 11 cars.   (*c*) 133.6 H. P., nearly.

(1533)   What three most important points must be kept in view in the construction of tail-rope couplings?

(1534)   What are the uses of the couplings of the tail-rope for a main and tail rope haulage?

(1535)   Describe in your own words what you consider the best kind of main socket for main and tail ropes.

(1536)   Describe a slip or detaching-hook for disconnecting a haulage-rope with a train of cars.

(1537)   Why is it sometimes better to locate the engine and drum for underground haulage at the surface than to fix them in the mine?

(1538)   Describe the modes of conducting the haulage-ropes from the surface to the underground workings of a mine.

(1539)   Which are better, double or single tracks for main and tail rope haulage in the underground workings of a mine?

(1540)   Describe the principles of action of the endless-rope system of haulage.

(1541)   Explain the arrangement of the rope band in reference to three of its most important features in endless-rope haulage.

(1542)   How are the ropes kept tight in an endless-rope haulage?

(1543)   Explain an endless-rope haulage with a single band.

(1544)   Explain how a single endless-rope band may be made to do the haulage out of several branching districts, and include in your description the great defects in such an arrangement.

(1545)   Explain how an endless-rope haulage is done with main bands and branching bands.

(1546)   Show the advantage of gravity-planes worked with an endless rope over those of the other systems of double ropes.

(1547)   Show why haulage with an endless-rope system is cheaper for undulating roads than by main and tail rope and other systems of haulage.

(1548)   Show the advantage of working each of the deflecting bands of an endless-rope haulage with motors that are actuated by transmitted energy.

(1549)   Show how a gasoline-engine might be applied for bands of district haulages.

(1550)   Show that in large mines, where the coal has to be hauled out of many districts, it can be better and more cheaply

done in many cases by the endless-rope system than by any other known system of haulage.

(1551) Show that locomotive haulage is better in small mines than large ones, and especially where the roads are comparatively level.

(1552) Show in what important respect the fixing of a tail-sheave for a tail-rope differs from the tail-sheave for an endless-rope haulage.

(1553) What should be the smallest diameter of a sheave to prevent damage from the bending of a steel rope having 19 wires to the strand ?

(1554) What should be the smallest diameter of a sheave to prevent damage from the bending of a steel rope having 7 wires to the strand ?

(1555) Explain two good tension arrangements for keeping an endless rope tight.

(1556) Explain two methods of attaching and detaching the cars of an endless-rope band.

(1557) The output of a certain mine is 2,500 tons per day, and the length of the road along which the main haulage-band runs the cars is 5,230 feet. A car carries 1½ long tons of coal. The velocity of the rope is 2 miles an hour, and the time for one day's running is 10 hours. As the rope is sometimes stopped to wait for work, the average running time is only 8 hours. (a) How many loaded cars are there on the haulage-band at one time ? (b) What are their distances apart ?   Ans. $\begin{cases} (a) & 103 \text{ cars.} \\ (b) & 50.68 \text{ ft.} \end{cases}$

(1558) The output in tons, the velocity of the rope, and the weight of coal a car carries are unaltered. If the length of the haulage-road is made one-half of the original length, how will the distances apart of the cars be affected ?

(1559) How much can I reduce the weight of a rope by doubling its velocity, and haul out the same weight of coal per day as was hauled before with a heavy rope running at a lower velocity ?

(1560) Which is the best speed at which to run an endless-rope haulage, two or four miles an hour ?

(1561) How are the ropes attached to the cars in an end-less-rope haulage ?

(1562) The track for a single band in an endless-rope haulage is 4,720 feet in length, and this rope hauls out 976 long tons of coal in 10 hours. The cars carry 1¼ long tons of coal, and an empty car weighs 1,200 pounds. The velocity of the rope is 2½ miles an hour, and the weight of the rope is 3 pounds per foot of length. What is (a) the tension in the rope, and (b) the horsepower of the hauling engine, sup-posing the mean grade of the road to be a true level ?

Ans. $\begin{cases} (a) & 4337.47 \text{ lb.} \\ (b) & 28.92 \text{ H. P.} \end{cases}$

(1563) The track for a single band on an endless-rope haulage is 4,720 feet in length, and the rope band hauls out 976 long tons of coal in 10 hours. The cars carry 1¼ long tons of coal, and an empty car weighs 1,200 pounds. The velocity of the rope is 2½ miles an hour, the weight of the rope is 3 pounds per foot of length, and the road has a mean up grade to the shaft of 2½ per cent. What is (a) the tension in the rope, and (b) the horsepower of the hauling engine ?

Ans. $\begin{cases} (a) & 6291.8 \text{ lb.} \\ (b) & 41.9 \text{ H. P.} \end{cases}$

(1564) The track for a single band on an endless-rope haulage is 4,720 feet in length, and the rope band hauls out 976 long tons of coal in 10 hours. The cars carry 1¼ long tons of coal, and an empty car weighs 1,200 pounds. The velocity of the rope is 2½ miles an hour, the weight of the rope is 3 pounds per foot, and the road has a mean down grade to the shaft of 2½ per cent. What is (a) the tension in the rope, and (b) the horsepower of the hauling engine ?

Ans. $\begin{cases} (a) & 2,383.14 \text{ lb.} \\ (b) & 15.89 \text{ H. P.} \end{cases}$

(1565) What is the number of strands in a wire rope for mine haulage, and between what numbers do the wires in a strand vary ?

(1566) Explain the difference between a twisted and a locked wire rope.

(1567) Show in what way the durability of a wire rope is affected by bending around and over sheaves.

(1568) Explain, without going much into detail, the general principles that govern the splicing of a rope.

(1569) Explain how a roadbed should be made for underground haulage.

(1570) Briefly explain three different kinds of rollers used for supporting the haulage-rope.

(1571) Briefly explain the mode of attachment and the mode of action of the two classes of engines used for underground haulage in mines.

(1572) As drums can not be used for endless-rope haulage, explain the construction and mode of action of grip-wheels used for that purpose.

(1573) Name the four types of locomotives that are in use for underground haulage in mines.

(1574) If a steam locomotive is used for underground haulage in a mine, on what roads should it run, and what should be the condition of the air in the passage in which the engine runs?

(1575) What particular plant is required for a compressed-air locomotive haulage?

(1576) What should be done in the pumping of compressed air to prevent waste of energy?

(1577) What difference should there be between the pressure of the air in a locomotive-tank and that of the pipe-line?

(1578) What special construction is provided for the night and day work of air-compressors?

(1579) How does temperature affect the pressure of the compressed air in the pipe-lines provided for locomotive haulage in mines?

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(1580) What is the advantage of a compressed-air loco-
motive over that of a steam-power locomotive for under-
ground haulage in mines ?

(1581) What difficulty would occur in using gasoline
locomotive-engines for underground haulage in mines?

(1582) Show in what important respect the grades are
different for locomotive and for rope haulage in mines.

(1583) What rails require elevating on curves for loco-
motive and for rope haulage ?

(1584) What important feature is it necessary for the
mining engineer or mine manager to understand in the
construction of self-acting inclines ?

A KEY

TO ALL THE

QUESTIONS AND EXAMPLES

CONTAINED IN THE

EXAMINATION QUESTIONS

INCLUDED IN THIS VOLUME.

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The Keys that follow have been divided into sections cor-
responding to the Examination Questions to which they
ref·r, and have been given corresponding section numbers.
The answers and solutions have been numbered to corre-
spond with the questions. When the answer to a question
involves a repetition of statements given in the Instruction
Paper, the reader has been referred to a numbered article,
the reading of which will enable him to answer the question
himself.

To be of the greatest benefit, the Keys should be used
sparingly. They should be used much in the same manner
as a pupil would go to a teacher for instruction with regard
to answering some example he was unable to solve. If used
in this manner, the Keys will be of great help and assist-
ance to the student, and will be a source of encouragement
to him in studying the various papers composing the Course.

MECHANICS.

(PART 1.)

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(1) See Arts. **1800** and **1801.**

(2) (*a*) Applying formula **102,**

$$D = \frac{1\frac{1}{2} \times 50}{3.1416} = 23.87 \text{ in. Ans.}$$

(*b*) See Art. **1872.** Addendum = .3 of the pitch. 1.5 in. × .3 = .45 in. .45 in. × 2 = .9 in. = difference between the diameter of the pitch circle and the outside diameter. Hence, outside diameter = 23.87 + .9 = 24.77 in. Ans

(3) Apply formula **110.**
Pitch = $\frac{1}{13}$ in.; therefore,

$$W = \frac{6.2832 \times 24 \times 11}{\frac{1}{13}} = 21,563.94 \text{ lb. Ans.}$$

(4) The pull on the support equals the centrifugal force of the ball. Hence, applying formula **112,**

$$F = .00034 \times 5 \times 3\frac{2}{13} \times 350^2 = 555\frac{1}{3} \text{ lb. Ans.}$$

(5) Apply formula **113.**

$$K = \frac{2 \times 600^2}{64.32} = 11,194 \text{ ft.-lb. Ans.}$$

(6) 7 ft. = 84 in. Arc of contact = $\dfrac{84}{63 \times 3.1416} \times 360°$
= 153°. 800 + 3 (180 − 153) = 881. Applying formula **115,**

$$W' = \frac{881 \times 150}{3,000} = 44.05 \text{ in.}$$

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Using formula **117**,

$$W_1 = 44.05 \times \tfrac{2}{3} = 29.37 \text{ in., or say } 29.5 \text{ in.} \quad \text{Ans.}$$

(**7**) See Arts. **1802** to **1823**.

(**8**) See Art. **1840**.

(**9**) See Arts. **1860** and **1861**.

(**10**) 13 ft. = 156 in. Applying formula **99**,

$$N = \frac{91 \times 108}{156} = 63 \text{ rev. per min., the speed of the engine.}$$
Ans.

(**11**) See Art. **1899**.

(**12**) Arc of contact $= \dfrac{18}{14 \times 3.1416} \times 360° = 147°$. 800

$+ 3 (180 - 147) = 899$. Applying formula **115**,

$$W = \frac{899 \times 2.5}{2,000} = 1.12 \text{ in., say 1 in.} \quad \text{Ans.}$$

(**13**) (*a*) See Arts. **1810** and **1835**.

(*b*) and (*c*) See Art. **1809**.

(**14**) $\tfrac{48}{12} \times 3.1416 = 12.5664$ ft. = circumference of pulley. $\dfrac{3,000}{12.5664} = 238.73$ revolutions in 1 minute, or 60 seconds. To make 100 revolutions will require $\dfrac{100}{238.73} \times 60 = 25.13$ sec., nearly. Ans.

(**15**) 4 ft. 6 in. = 54 in. $54 \times 2 \times \tfrac{3}{4} \times .261 = 21.141$ lb. = weight of lever. Considering the weight of the lever to be concentrated at its center of gravity, we have three weights of 47, 21.141, and 71 lb., with the smaller weight $\tfrac{54}{2} = 27$ in. from the other two. To find the center of gravity of the two large weights, apply formula **93**. $l_1 = \dfrac{47 \times 54}{71 + 47} = 21.508$ in. = the distance bc in Fig. I. Consider both weights to be concentrated at b; that is, imagine both weights removed and to be replaced by the dotted

weight W, equal to $71 + 47 = 118$ lb. The dotted circle w represents the weight of the bar. The distance $ae = 27$
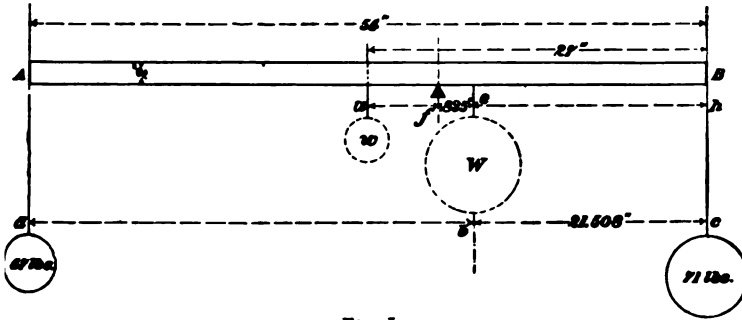
FIG. I.

$- 21.508 = 5.492$ in. Distance of balancing point f from e is found by means of formula **93** to be $\dfrac{21.141 \times 5.492}{118 + 21.141}$ $= .835$ in. Finally, $fh =$

$$21.508 + .835 = 22.343 \text{ in.} = \text{short arm.} \quad \text{Ans.}$$
$$54 - 22.343 = 31.657 \text{ in.} = \text{long arm.} \quad \text{Ans.}$$

(16) See Art. **1899.** $\dfrac{51}{62.5} = .816$, the specific gravity.

                                                      Ans.

(17) See Arts. **1824** and **1826.**

(18) See Art. **1830.**

(19) In Fig. II, ABC represents the triangle. The center of gravity is found as explained in Art. **1845.** The distance of the center of gravity from the side $AC = 1\frac{3}{32}$ in.   Ans.
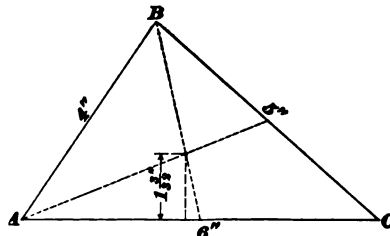
FIG. II.

(20) Speed of a point on the pitch circle in feet per minute $= \frac{30}{12} \times 3.1416 \times 100 = 785.4$ ft. per min. Apply formula **109.**

     H. P. $= .01 \times 785.4 \times 1.57^2 = 19.36$.   Ans.

(21) See Art. **1826.**

(22) Applying formula **100,**

$$P = \frac{6,000 \times 6 \times 5 \times 8 \times 3}{18 \times 12 \times 15 \times 12} = 111\tfrac{1}{9} \text{ lb.}$$

Since there is a loss of 20%, $111\tfrac{1}{9}$ represents 80% of the total force. Hence, the force actually required $= 111\tfrac{1}{9} \div$ $80 = 138\tfrac{8}{9}$ lb.   Ans.

(23) Apply formula **104.**

$$P = \frac{3.1416 \times 24.16}{38} = 1.9974 \text{ in.}   \text{Ans.}$$

(24) See Art. **1858.** Since there are eight parts of the rope, the force required $= 1,890 \div 8 = 236\tfrac{1}{4}$ lb.   Ans.

(25) Volume $= (\tfrac{1}{2})^3 \times .7854 \times 10 = 1.963$ cu. in.   One cu. in. of lead weighs .411 lb. (see table of Weights per Cubic Foot); consequently, $1.963 \times .411 = .807$ lb. $= 12.91$ oz.
Ans.

(26) Length of power-arm $= 4$ ft. $- 4$ in. $= 48$ in. $- 4$ in. $= 44$ in.   According to formula **94,** $P \times 44 = 1,500$ $\times 4 = 6,000$; hence, $P = \dfrac{6,000}{44} = 136\tfrac{4}{11}$ lb.   Ans.

(27) Length of power-arm $= 4$ ft. $= 48$ in.   Hence, as in the preceding question, $P = \dfrac{6,000}{48} = 125$ lb.   Ans.

(28) See Arts. **1869** and **1870.**

(29) See Art. **1885.** $4,000 \times 45 = 400 \times$ the force. Hence, force $= \dfrac{4,000 \times 45}{400} = 450$ lb.   Ans.

(30) 14 ft. $= 168$ in.   Applying formula **114,**

$$B = 3\tfrac{1}{4} \times \frac{18 + 14}{2} + 2 \times 168 = 388 \text{ in.} = 32 \text{ ft. } 4 \text{ in.}   \text{Ans.}$$

(31) See Arts. **1871** and **1872.**

(32) See Arts. **1876** and **1877.**

(33)   The weight which comes on the block and tackle is the same as the force required to pull the body up the plane, or is equal to $\dfrac{50,000 \times 125}{1,200} = 5,208\frac{1}{3}$ lb. Since there are 12 parts to the rope, the force required to be exerted on the free end is $5,208 \div 12 = 434$ lb.   Ans.

(34)   See Art. **1898.**

(35)   See Art. **1838.**

(36)   Applying formula **95,** letting P represent the required force,

$$P \times 30 \times 20 \times 10 \times 15 = 1,250 \times 6 \times 5 \times 4 \times 7,$$

or $P = \dfrac{1,250 \times 6 \times 5 \times 4 \times 7}{30 \times 20 \times 10 \times 15} = 11\frac{2}{3}$ lb.   Ans.

(37)   See Art. **1872.**

(38)   One cubic foot of water weighs 62.5 lb.; hence, 20 cu. ft. weigh $62.5 \times 20 = 1,250$ lb. The work done $= 1,250 \times 50 = 62,500$ ft.-lb.   Ans.

(39) $18,000 + 10,000 = 28,000$ lb. $=$ the load which the screw must overcome.

Using formula **111,**

$$P = \frac{\frac{1}{2} \times 28,000}{6.2832 \times 15} = 99 \text{ lb., nearly.} \quad \text{Ans.}$$

(40) $30 \times 14\frac{1}{2} \times 2 = 870.$ $870 \div 5 = 174$ lb.   Ans.
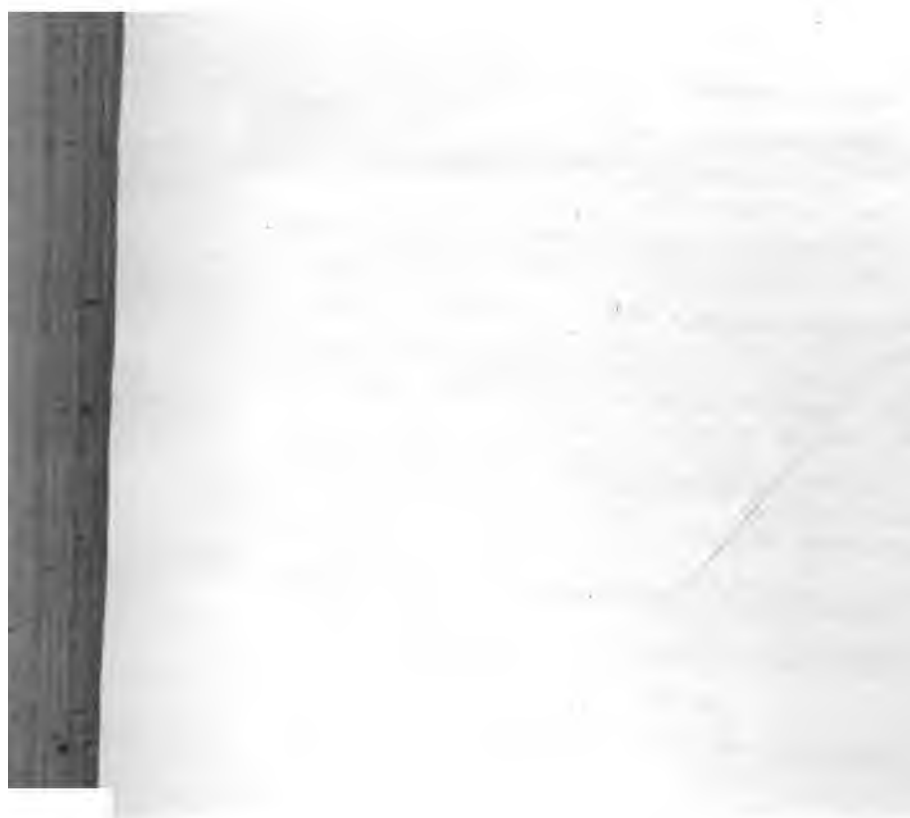
MECHANICS.

(PART 2.)

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(1) That force which will produce the same final effect upon a body as all the other forces acting separately or together.

(2) This example is solved by the parallelogram of forces, as in Art. **1917.** Measuring the diagonal, the total pressure on the shaft is found to be 7¼ tons, nearly.   Ans.

(3) See Arts. **1932** and **1933.**

(4) Applying formula **122,**

$$W = 12,000 \times (\tfrac{3}{8})^2 = 1,687.5 \text{ lb.}   \text{ Ans.}$$

(5) Apply formula **125,** and use 1,000 instead of 600, as the rope is of steel.

$$W = 1,000 \times (5\tfrac{1}{4})^2 = 27,562.5 \text{ lb.}   \text{ Ans.}$$

(6) (*a*) If a 5-inch line = 20 lb., a 1-inch line = 4 lb. 1 ÷ 4 = ¼ inch = 1 lb.   Ans.

(*b*) 6¼ ÷ 4 = 1.5625 inches = 6¼ lb.   Ans.

(7) See Art. **1964.**

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MECHANICS.

(PART 2.)

(1) That force which will produce the same final effect upon a body as all the other forces acting separately or together.

(2) This example is solved by the parallelogram of forces, as in Art. **1917.** Measuring the diagonal, the total pressure on the shaft is found to be $7\frac{1}{4}$ tons, nearly.   Ans.

(3) See Arts. **1932** and **1933.**

(4) Applying formula **122,**

$$W = 12,000 \times (\tfrac{3}{8})^2 = 1,687.5 \text{ lb.}\quad \text{Ans.}$$

(5) Apply formula **125,** and use 1,000 instead of 600, as the rope is of steel.

$$W = 1,000 \times (5\tfrac{1}{4})^2 = 27,562.5 \text{ lb.}\quad \text{Ans.}$$

(6) (*a*) If a 5-inch line $= 20$ lb., a 1-inch line $= 4$ lb. $1 \div 4 = \frac{1}{4}$ inch $= 1$ lb.   Ans.

(*b*) $6\frac{1}{4} \div 4 = 1.5625$ inches $= 6\frac{1}{4}$ lb.   Ans.

(7) See Art. **1964.**

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(8) The method of obtaining the resultant is shown in Fig. I. The forces are laid off to scale to form a polygon, and the closing line gives the direction and magnitude of the resultant. See Art. **1918.**
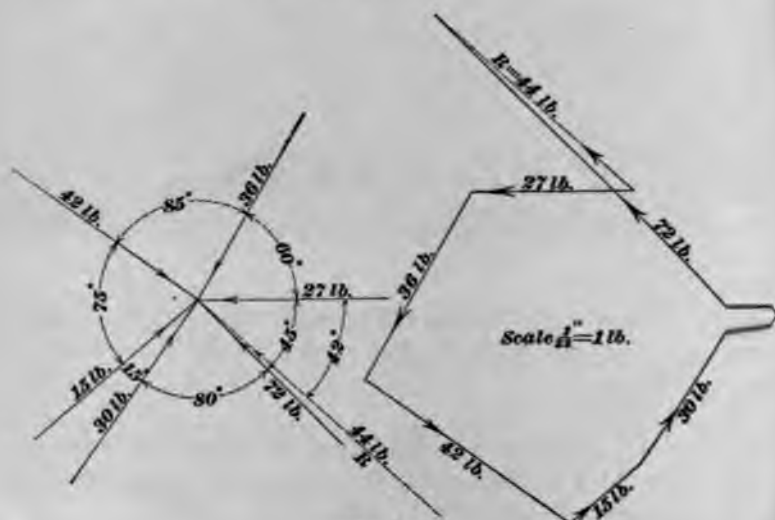
FIG. I.

(9) Area of cross-section $= 8^2 \times .7854 = 50.2656$ sq. in. 10 ft. $= 120$ in. $= L$. Crushing strength $= 3.5$ tons per sq. in. (see Table 33). $a = 187.5$ (see Table 36). Substituting these values in formula **127,**

$$W = \frac{3.5 \times 50.2656}{\dfrac{120^2}{187.5 \times 8^2} + 1} = 80 \text{ tons, very nearly.}$$

Hence, $80 \div 6 = 13\frac{1}{3}$ tons $=$ safe load. Ans.

(10) Those forces by which the given force may be replaced and which will produce the same effect on a body as the given force.

(11) Apply formula **119.**

$$A = \frac{12,000}{5,000} = 2.4 \text{ sq. in., the area of the bolt.}$$

$$\text{Diameter} = \sqrt{\frac{2.4}{.7854}} = 1.74+ \text{ in.}\quad \text{Ans.}$$

(12)　First calculate the load it will sustain in the middle, by means of formula **130.**

$$\text{Load in middle} = \frac{4 \times 10^3 \times 8 \times 30}{28} = 3,428\tfrac{4}{7} \text{ lb.}$$

$$\text{Uniform load} = 3,428\tfrac{4}{7} \times 2 = 6,857\tfrac{1}{7} \text{ lb.}\quad \text{Ans.}$$

(13)　Apply formula **133.**　From Table 40, the proper constant is 70.

$$\text{Horsepower} = \frac{10^3 \times 200}{70} = 2,857\tfrac{1}{7}.\quad \text{Ans.}$$
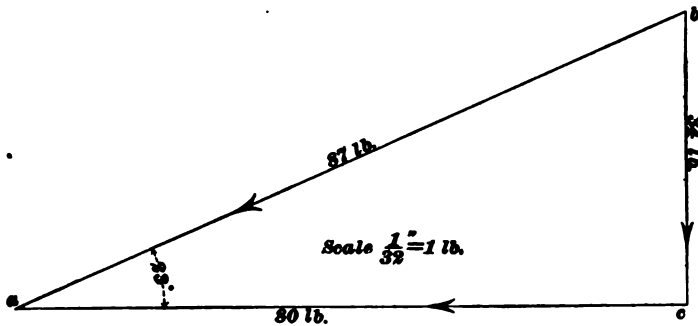
Fig. II.

(14)　See Fig. II.　By trigonometry, $bc = 87 \times \sin 23°$ $= 87 \times .39073 = 33.994$ lb.　$ac = 87 \times \cos 23° = 87 \times .92050$ $= 80.084$ lb.

(15)　Apply formula **121.**

$$W = 18,000 \times .5^2 = 4,500 \text{ lb., the load.}\quad \text{Ans.}$$

(16)　Applying formula **126,**

$C = .0408 \sqrt{14,000} = 4.83$ in., the circumference, nearly.　Ans

(17) See Fig. III.

Scale $\frac{1''}{8} = 1$ lb.

Fɪɢ. III.

(18) See Fig. IV.

Scale $\frac{1''}{8} = 1$ lb.
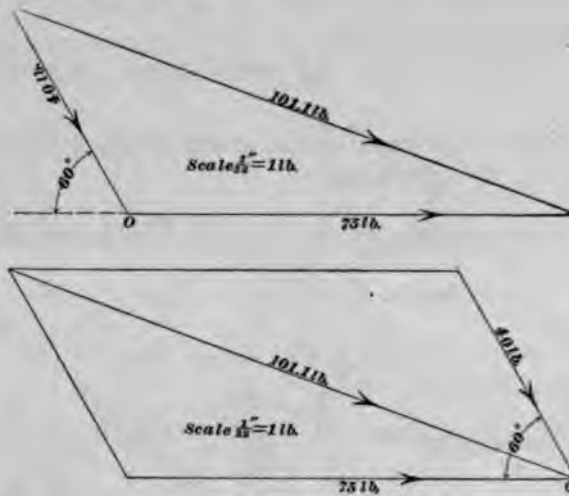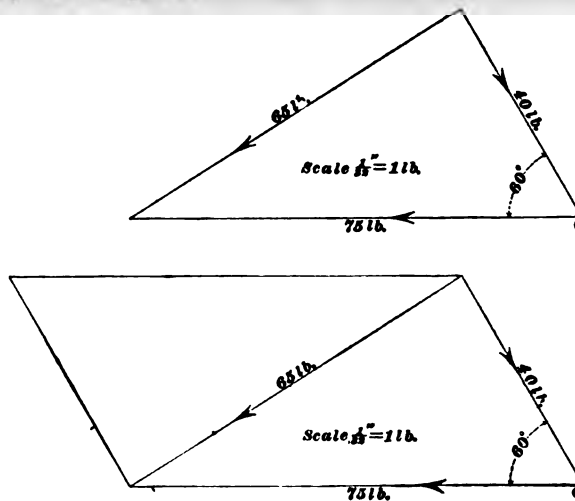
Fɪɢ. IV

(19) $46 - 27 = 19$ lb., acting in the direction of the force of 46 lb. Ans.

(20) Area of cross-section = $1\frac{3}{4} \times 3 = 5.25$ sq. in. Applying formula **118,**

$$W = 5.25 \times 6,000 = 31,500 \text{ lb., the safe load.}\quad \text{Ans.}$$

(21) The graphical construction is clearly shown in Fig. V.
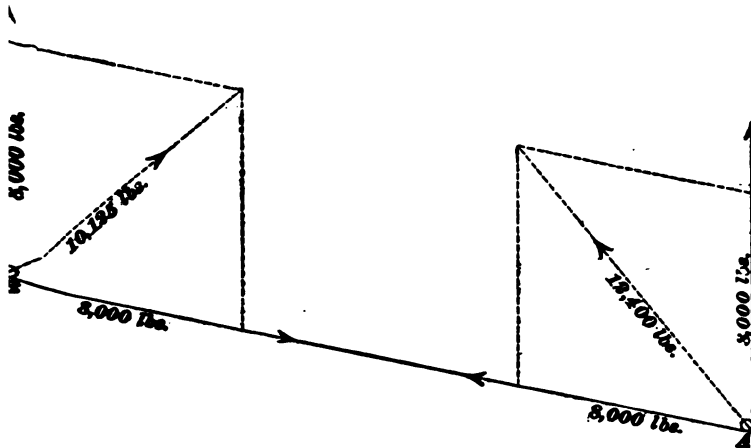
FIG. V.

(22) See Arts. **1926** to **1928.**

(23) Apply formula **126,** and use .0316 instead of .0408, since the rope is of steel.

$$C = .0316 \sqrt{8,000} = 2.83 \text{ in.}\quad \text{Ans.}$$

(24) Apply formula **130,** and multiply the result by 2.

$$W = \frac{4 \times 6^2 \times 2 \times 160}{20} \times 2 = 4,608 \text{ lb., the load.}\quad \text{Ans.}$$

(25) See Arts. **1929** to **1931.**

(26) Apply formula **125.**

$$W = 600 \times 6^2 = 21,600 \text{ lb.}\quad \text{Ans.}$$

24—38

(27) 4 ft. = 48 in. Area to be sheared = $48 \times \frac{1}{4}$ = 24 sq. in. Applying formula **132,**

$W = 24 \times 40,000 = 960,000$ lb., the force required. Ans.

(28) Applying formula **134,**

$\dfrac{70 \times 200}{7^2} = 40.8$ revolutions per minute, nearly. Ans.

(29) Apply formula **125.**

Load $= 600 \times 4^2 = 9,600$ lb. Ans.

(30) Apply formula **128.**

Load $= \dfrac{2.5^2 \times 1.5 \times 100}{4\frac{8}{12}} = 201$ lb., nearly. Ans.

(31) Apply formula **133.**

Horsepower $= \dfrac{(2\frac{7}{16})^3 \times 120}{85} = 20.445$ H. P. Ans.

(32) Area of cross-section $= (1\frac{1}{2})^2 \times .7854 = 1.7671$ sq. in Apply formula **118.**

Safe steady load $= 12,000 \times 1.7671 = 21,205.2$ lb. Ans.

(33) Substituting the values of $C = 40$, $S = 14^2 \times .7854$ $- 11.5^2 \times .7854 = 50.0693$, $L = 20 \times 12 = 240$, $a = 562.5$, and $d = 14$ in formula **127,** we have

$$W = \dfrac{40 \times 50.0693}{\dfrac{240^2}{562.5 \times 14^2} + 1} = \dfrac{2,002.772}{1.5225} = 1,315.45 \text{ tons.}$$

$$\dfrac{1,315.45}{6} = 219.24 \text{ tons.} \quad \textbf{Ans.}$$

(34) See Art. **1963.** Area punched $= 1\frac{1}{2} \times 3.1416$ $\times \frac{3}{4} = 3.5343$ sq. in. Force $= 3.5343 \times 60,000 = 212,058$ lb.
                                                                                    Ans.

(35)   See Fig. VI
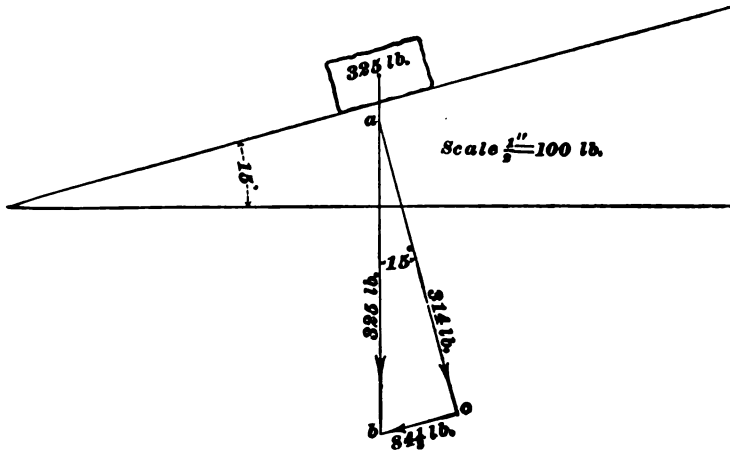
FIG. VI.

(*a*) $a\,c = 325 \times \cos 15° = 325 \times .96593 = 313.93$ lb.   Ans.

(*b*) $b\,c = 325 \times \sin 15° = 325 \times .25882 = 84.12$ lb.   Ans.

STEAM AND STEAM-BOILERS.

(1188) See Arts. **1970** to **1977.**

(1189) See Arts. **1972** to **1974.**

(1190) Less. See Art. **1982.**

(1191) See Arts. **1975** to **1977.**

(1192) (*a*) See Art. **1979.**

   (*b*) and (*c*) See Art. **1978.**

   (*d*) See Art. **1982.**

(1193) See Arts. **1984** to **1986.**

(1194) (*a*) and (*b*) See Arts. **1980** and **1981.**

   (*c*) 1 B. T. U. = 778 ft.-lb.

   30½ B. T. U. = 30½ × 778 = 23,729 ft.-lb. **Ans**

(1195) 35 H. P. = 35 × 33,000 ft.-lb. per min. = 35 × 33,000 × 60 ft.-lb. per hour = $\dfrac{35 \times 33,000 \times 60}{778}$ B. T. U. per hour = 89,074.5 B. T. U. per hour.

But this is the heat actually used, or 20% of the whole. Hence, the heat required is 89,074.5 ÷ .20 = 445,372.5 B. T. U. per hour. Ans.

(1196) One horsepower = 33,000 × 60 ft.-lb. per hour = $\dfrac{33,000 \times 60}{778}$ B. T. U. per hour.

Each pound of coal gives 14,000 B. T. U., of which 8%, or 14,000 × .08 = 1,120 B. T. U., is utilized. Hence, the coal required per hour per H. P. is

$$\dfrac{33,000 \times 60}{778} \div 1,120 = 2.27 \text{ lb.} \textbf{Ans.}$$

§ 18

(1197) The specific heat of sulphur is .2026. (See Table 41.) By formula **136,** $U = c\,W(t_1 - t) = .2026 \times 22\frac{1}{2} \times (68 - 44) = 109.4$ B. T. U. Ans.

(1198) (*a*) See Art. **1984.**

(*b*) To raise the ice from 17° to 32° requires for each pound $.504 \times (32 - 17) = 7.56$ B. T. U. To melt it requires 144 B. T. U. Hence, 1 lb. requires $144 + 7.56 = 151.56$ B. T. U. 11 lb. requires $11 \times 151.56 = 1{,}667.16$ B. T. U.

Ans.

(1199) By formula **136,** $U = c\,W(t_1 - t) = .4805 \times 6 \times (342 - 310) = 92.256$ B. T. U. Ans.

(1200) Using formula **137,**

$$T = \frac{w\,c\,t + w_1 c_1 t_1 + w_2 c_2 t_2 + \cdots}{w\,c + w_1 c_1 + w_2 c_2 + \cdots} =$$

$$\frac{18 \times .0951 \times 305 + 13 \times .1138 \times 278 + 32 \times 1 \times 56}{18 \times .0951 + 13 \times .1138 + 32 \times 1} = 77.45°. \quad \text{Ans.}$$

(1201) (*a*) 966 B. T. U. Ans.

(*b*) To raise a pound of water from 63° to 212° requires $212 - 63 = 149$ B. T. U. To change it into steam requires 966.069 more B. T. U. $966 + 149 = 1{,}115$ B. T. U. for 1 lb. Hence, $8 \times 1{,}115 = 8{,}920$ B. T. U. are required. Ans.

(1202) To change 1 lb. of ice from 23° to 32° requires $(32 - 23) \times .504 = 4.536$ B. T. U. To melt the ice requires 144 B. T. U. To change the water at 32° to water at 212° requires 180 B. T. U. per pound. To change the water at 212° into steam at 212° requires 966 B. T. U. per pound. $4.536 + 144 + 180 + 966 = 1{,}294.536$ B. T. U. per pound. For 2.2 pounds, $1{,}294.536 \times 2.2 = 2{,}847.98$ B. T. U., as required. Ans.

(1203) See Arts. **1991** to **1993.**

(1204) See Art. **2024** and Arts. **2027** to **2030.**

(1205) In the return-tubular boiler the one or two large flues are replaced by a large number of small tubes. In other respects, the boilers are quite similar in principle.

(1206) See Art. **2024.**

(1207) See Arts. **2017, 2023, 2025,** and **2028.**

(1208) See Art. **2024.**

(1209) See Art. **2023.**

(1210) See Art. **2011.**

(1211) See Art. **2011.**

(1212) (*a*) See Art. **2004.**

(*b*) The temperature at which combustion takes place is always the same for the same substance. The nitrogen reduces the temperature of the furnace, since a portion of the heat given off by combustion is required to heat the nitrogen.

(1213) No. See definition of combustion, Art. **2003.**

(1214) See Art. **2007.**

(1215) See Art. **2008.**

(1216) The number of heat units required to convert a pound of water at 32° into steam at 400° may be found by means of formula **140.**

$$H = 1,081.4 + .305 \times 400 = 1,203.4 \text{ B. T. U.}\quad \text{Ans.}$$

(1217) In order to use formula **140,** the temperature must be known. This can be found when the pressure is known, by means of formula **138.** Applying the formula, $t = 14 \sqrt{175} + 199 = 384.2°$, the temperature of saturated steam having a pressure of 175 pounds per square inch. Now, using formula **140,**

$$H = 1,081.4 + .305 \times 384.2 = 1,198.6 \text{ B. T. U.}\quad \text{Ans.}$$

(1218) Since the expansion follows Mariotte's law, the final pressure may be found by the formula $p_1 = \dfrac{p\,v}{v_1}$. Substituting, $p_1 = \dfrac{60 \times 5}{5 \times 2.5} = 24$ lb. per sq. in. above vacuum. $24 - 14.7 = 9.3$ lb. per sq. in. above atmosphere. **Ans.**

(1219) From Table 42, column 5, the total heat of combustion of one pound of coal is found to be 14,133 B. T. U.

11 × 13 × 5 = 715 pounds of coal burned in 5 hours.

14,133 × 715 = 10,105,095 B. T. U. generated by the combustion of the coal. Ans.

(1220) According to Table 42, the amount of air required for the complete combustion under a blast draft is found to be 14 pounds. Hence, the amount of air required for combustion of the coal in Question 1219 is

715 × 14 = 10,010 pounds. Ans.

(1221) The number of pounds of water having a temperature of 62° which can be converted into steam having a temperature of 212° is found, from Table 42, column 6, to be 12.67 pounds. Hence, the total quantity of water which could be evaporated under the above conditions by the combustion of 715 pounds of coal is

12.67 × 715 = 9,059.05 pounds. Ans.

(1222) Since the pressure is 3,600 pounds per *square foot* above a vacuum, and there are 144 square inches in a square foot, the pressure above a vacuum is $\frac{3,600}{144} = 25$ pounds per *square inch*. Consequently, the pressure per square inch above the atmosphere is $25 - 14.7 = 10.3$ pounds. Ans.

(1223) See Art. **2001.**

(1224) According to formula **138,** the required temperature is

$$t = 199 + 14 \times \sqrt{152} = 371.62° \text{ F}. \quad \text{Ans.}$$

(1225) Applying formula **139,** we have for the required pressure $p = \left(\frac{232 - 199}{14}\right)^2 = 5.56$ pounds per square inch gauge-pressure. Ans.

(1226) 132 tons equal 132 × 2,000 = 264,000 pounds. 264,000 × 296 = 78,144,000 = foot-pounds of work necessary

to raise the coal to the top of the shaft.   Since 1 B. T. U. =
778 foot-pounds, the heat supplied is

$$\frac{78,144,000}{778} = 100,442.15 \text{ B. T. U.}\quad \text{Ans.}$$

(1227) $277,160 \times 778 = 215,630,480$ foot-pounds of work
done by the engine in two hours.

Hence, $\dfrac{215,630,480}{2} = 107,815,240$ ft.-lb. done in one hour.
                                                              Ans.

(1228)   The strength of any construction is always that
of its weakest part.   In the present example the diameter
and thickness of the steam and water drums only are given,
the thickness of the flues, mud-drum, and boiler-shell, and the
diameter of the boiler-shell being omitted.   Such being the
case, we must confine ourselves to the strength of the steam
and water drums, assuming that the other parts of the
boiler have been made correspondingly strong.   The pres-
sure which the steam-drum will safely sustain is found by

formula **141** to be $\dfrac{16,608 \times \frac{5}{16}}{24} = 216.25$ pounds per square

inch, and the pressure which the water-drum will safely sustain

is found by the same formula to be $\dfrac{16,608 \times \frac{5}{16}}{20} = 259.5$ pounds

per square inch.   Since the safe pressure upon the steam-
drum is less than that which can be sustained by the water-
drum, the pressure on the boiler must not exceed the safe
pressure which can be sustained by the steam-drum; that is,
216.25 pounds per square inch.   Ans.

(1229)   From Table 43, it is seen that from 14 to 18
square feet of water-heating surface are required to produce
one horsepower with a return-tubular boiler.   Using 16
square feet as a mean, we obtain

$$\frac{1,620}{16} = 101\tfrac{1}{4} \text{ H. P.}\quad \text{Ans.}$$

(1230)   In the same manner as in example 1229, it is
found that about 11 square feet of heating-surface are

required to produce 1 horsepower with a water-tube boiler. Hence,

$$H. P. = \frac{3,025}{11} = 275 \text{ horsepower. Ans.}$$

(**1231**) Applying formula **142,** the height of the chimney is found to be

$$k = \left[\frac{348}{3.33 \times 12 - 2\sqrt{12}}\right]^2 = \left[\frac{348}{3.33 \times 12 - (2 \times 3.464)}\right]^2 = 111 \text{ ft. Ans.}$$

(**1232**) The dome and the dry-pipe. See Arts. **2022** and **2023.**

(**1233**) See Art. **2019.**

(**1234**) Blow-off pipes are provided to remove the collected sediment. The boiler is also provided with manholes or handholes for cleaning purposes.

(**1235**) See Art. **2013.**

(**1236**) To avoid overheating and burning out the upper plates of the furnace. So long as the water is in contact with the plates which are next to the fire, they can not be overheated or burned.

(**1237**) See Art. **2023.**

(**1238**) See Art. **2017.**

(**1239**) See Art. **2018.**

(**1240**) Answer from the result of your own observations.

(**1241**) The steam-pipe conveys the steam after it is generated from the boiler to the place where it is used. The feed-water pipe is the one through which the water is introduced to the boiler. A blow-off pipe is one attached to the lower part of the boiler or to a mud-drum. It is used to empty the boiler of the whole or a part of its contents.

(**1242**) See Art. **2012.**

(**1243**) The arm of the safety-valve is a lever in which the power is the total steam-pressure on the valve, $6 \times 81 = 486$ pounds. The power arm is 2 inches, and the weight is 54 pounds. Calling the weight arm b, we have, from formula **94**,

$$Pa = Wb, \text{ or } 486 \times 2 = 54 \times b.$$

Hence, $b = \dfrac{486 \times 2}{54} = 18$ in.   Ans.

(**1244**) According to formula **141**, $p = \dfrac{10,224 \times \frac{3}{8}}{30} = 127.8$ pounds per square inch, the greatest pressure under which it would be safe to operate a boiler of these dimensions.

(**1245**) See Art. **2033.**

(**1246**) (*a*) See Art. **2035.**

(*b*) The top and sides of the furnace, and the tubes.

(**1247**) Using formula **141**, $p = \dfrac{13,152 \times \frac{5}{16}}{45} = 91\frac{1}{3}$ lb. per sq. in., the safe working pressure. Therefore it would be unsafe to use 110 lb. pressure.

(**1248**) (*b*) According to Table 43, a vertical boiler has from 15 to 20 square feet of heating-surface per horsepower. Assuming 18 sq. ft. per H. P., the heating-surface will be $35 \times 18 = 630$ square feet.   Ans.

(*a*) Since the heating-surface is 25 to 30 times the grate area, the latter must lie between $\frac{630}{25} = 25.2$ sq. ft. and $\frac{630}{30} = 21$ sq. ft. ; say about 23 sq. ft.   Ans.

(*c*) One horsepower is equivalent to an evaporation of 30 pounds of water per hour, the feed being at 100° and the steam-pressure at 70 pounds. The evaporation per hour is, therefore, $35 \times 30 = 1,050$ lb.   Ans.

STEAM-ENGINES.

(1249) The stationary parts of a plain slide-valve engine are the steam-cylinder, steam-chest, supply-pipe, exhaust-pipe, guide-bars, shaft-bearings, and the bed or frame of the engine.

(1250) The expansion curve of steam on an indicator-card represents the decrease of pressure of the steam after cut-off, corresponding to the increase of volume.

(1251) It passes its central position during the interval between the point of release of the steam from the head end of the cylinder, and the point of compression of the steam in the crank end of the cylinder, during the forward stroke of the piston, and conversely for the backward stroke.

(1252) Plain slide-valves usually cut off between one-half and full stroke.

(1253) The points of cut-off and release are marked, as shown in Figs. 46, 47, 48, and 49. The perpendicular distances from these points to the atmospheric line are measured. Multiplying the lengths of these perpendiculars by 45, the scale of the spring, we obtain

$$1.3750 \text{ in.} \times 45 = 61.8750 \text{ lb. for cut-off}$$
$$.5625 \text{ in.} \times 45 = 25.3125 \text{ lb. for release} \Big\}\ \text{Fig. 46.}$$

$$1.3750 \text{ in.} \times 45 = 61.8750 \text{ lb. for cut-off}$$
$$.6800 \text{ in.} \times 45 = 30.6000 \text{ lb. for release} \Big\}\ \text{Fig. 47.}$$

$$1.3800 \text{ in.} \times 45 = 62.1000 \text{ lb. for cut-off}$$
$$.1200 \text{ in.} \times 45 = 5.4000 \text{ lb. for release} \Big\}\ \text{Fig. 48.}$$

$$1.3700 \text{ in.} \times 45 = 61.650 \text{ lb. for cut-off}$$
$$.1200 \text{ in.} \times 45 = 5.4000 \text{ lb. for release} \Big\}\ \text{Fig. 49.}$$

§ 19

To find the back-pressure, in each case, find the perpendicular distance between the lowest point of diagram and
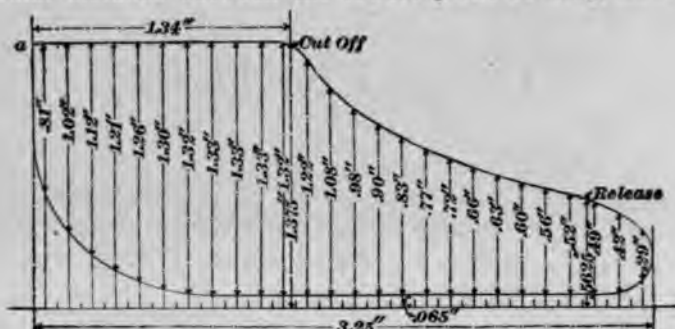
FIG. 46.
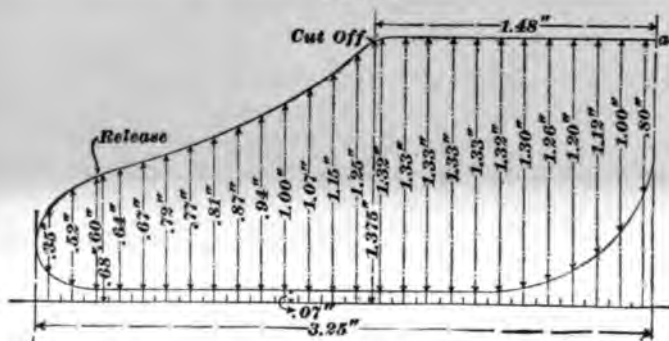
FIG. 47.
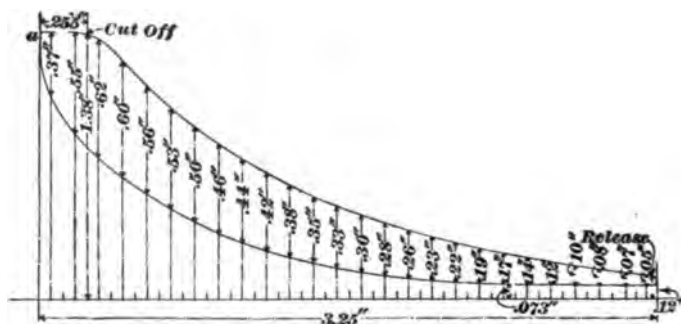
FIG. 48.

the atmospheric line; multiply by 45, the scale of the spring, and the products will be as follows:

.065 in. × 45 = 2.925 lb. back-pressure for Fig. 46.

.070 in. × 45 = 3.15   lb. back-pressure for Fig. 47.
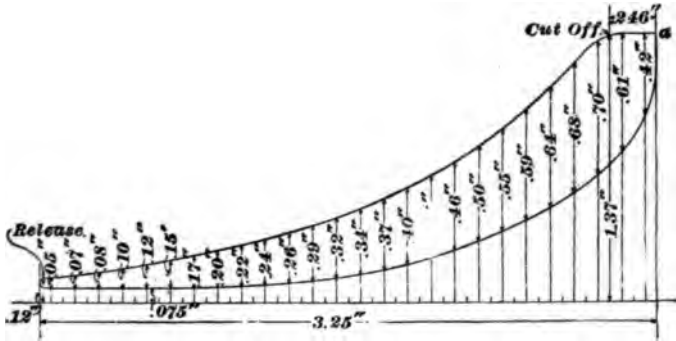
FIG. 49.
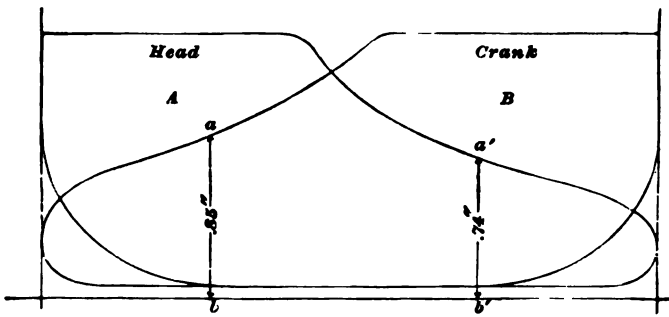
FIG. 50.
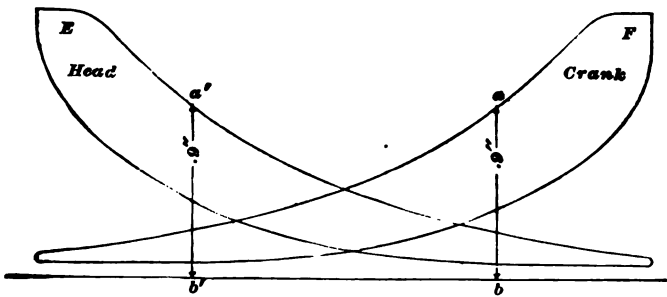
FIG. 51.

.073 in. × 45 = 3.285 lb. back-pressure for Fig. 48.

.075 in. × 45 = 3.375 lb. back-pressure for Fig. 49.

To determine the steam-pressure in the cylinder at the point of compression, we must combine diagrams A and B and E and F. These diagrams are combined by placing B upon A and F upon E, the atmospheric and extreme right and left hand lines coinciding. The height $a\,b$ of the diagram B, in Fig. 50, represents the pressure of the steam in the crank end of the cylinder at the point of compression of the diagram A. This is as it should be, since the compression curve is drawn by the pencil of the indicator when the piston is making its return stroke. In a similar manner, the pressure of the steam in the head end of the cylinder at the point of compression in the crank end is the height $a'\,b'$ of A. In Fig. 51 the height ab represents the pressure at compression for E, and $a'\,b'$ the same for F. These results tabulated are as follows:

.85 in. × 45 = 38.25 lb. for A.

.74 in. × 45 = 33.30 lb. for B.

.90 in. × 45 = 40.50 lb. for E.

.90 in. × 45 = 40.50 lb. for F.

(1254) Project the extreme right and left hand points of the indicator-diagrams upon the atmospheric line; divide the distance between them into any number of equal spaces—26 in this case—and through the centers of these spaces draw lines across the diagram perpendicular to the atmospheric line. Now measure the length in inches of each of these perpendicular lines (the lengths are given in all the figures), and take their sum; divide this sum by the number of the equal spaces into which the atmospheric line is divided, and multiply the quotient by the scale of the spring.

Sum of the perpendiculars of the diagram of Fig. 46 = 24.02 in.; then,

$$\frac{24.02}{26} \times 45 = 41.573 \text{ lb., M. E. P.}$$

Sum of the perpendiculars of the diagram, Fig. 47, = 26 in.; then,

$$\tfrac{26}{26} \times 45 = 45 \text{ lb., M. E. P.}$$

The average M. E. P. for both diagrams is

$$\frac{41.573 + 45}{2} = 43.29 \text{ lb. per sq. in. } \textbf{Ans.}$$

(1255) Sum of the perpendiculars of the diagram, Fig. 48, = 8.32 in.; then,

$$\frac{8.32}{26} \times 45 = 14.40 \text{ lb., M. E. P.}$$

Sum of the perpendiculars of the diagram, Fig. 4 = 8.97; then,

$$\frac{8.97}{26} \times 45 = 15.525 \text{ lb., M. E. P.}$$

The average M. E. P. for the two diagrams is

$$\frac{14.40 + 15.525}{2} = 14.96 \text{ lb. per sq. in. } \textbf{Ans.}$$

(1256) Area of 15-inch piston $= 15^2 \times .7854 = 176.71$ square inches.

Using formula **143,**

$$\text{I. H. P.} = \frac{43.29 \times 2 \times 176.715 \times 175}{33,000} = 81.14 \text{ I. H. P. } \textbf{Ans.}$$

(1257) Proceeding as in example 1256,

$$\text{I. H. P.} = \frac{14.96 \times 2 \times 176.715 \times 175}{33,000} = 28.04 \text{ I. H. P. } \textbf{Ans.}$$

(1258) The actual horsepower is $81.14 - 28.04 = 53.1$ H. P. **Ans.**

Applying formula **146,** the efficiency is

$$\frac{53.1}{81.14} = .654 = 65.4 \text{ per cent. } \textbf{Ans.}$$

(1259) The force of gravity and the centrifugal force.

(1260) See Art. **2098.**

(1261) The piston, piston-rod, cross-head, connecting-rod, crank, crank-shaft, eccentric, eccentric-rod, slide-valve, and fly-wheel.

(1262) In order that the energy stored in them may be utilized in carrying the crank over its dead-center position, and also to cause the engine to run at a more uniform speed.

(1263) Compression is taking place. See Figs. 50 and 51.

(1264) Any portion added to the length of a valve more than is absolutely necessary, in order to cover the outside edges of the steam-ports when the valve is in its central position, is called the outside lap of the valve. It is added to enable the valve to cut off the live steam before the piston reaches the end of its stroke.

(1265) Apply rule, Art. **2059.** Cut-off in the diagram, Fig. 807, takes place at a point 1.34 inches from *a*. See Fig. 46.

Therefore, cut-off equals $\dfrac{1.34}{3.25}$, or 41% of stroke.

Cut-off in the diagram, Fig. 808, takes place at a point 1.48 inches from *a*. See Fig. 47.

Therefore, cut-off equals $\dfrac{1.48}{3.25}$, or 46% of stroke, nearly.

Cut-off in the diagram, Fig. 809, takes place at a point .255 inch from *a*. See Fig. 48.

Therefore, cut-off equals $\dfrac{.255}{3.25}$, or 7.8% of stroke.

Cut-off in the diagram, Fig. 810, takes place at a point .246 inch from *a*. See Fig. 49.

Therefore, cut-off equals $\dfrac{.246}{3.25}$, or 7.6% of stroke.

In each case the length of the diagram is 3.25 inches.

(1266) The indicated horsepower of this engine will be about one-half greater than the actual horsepower, or $\frac{65}{2} + 65 = 97.5$ horsepower. See example, Art. **2077.** A fair piston speed is 500 feet per minute.

Assume the cut-off to be taken at $\frac{3}{4}$ and the boiler-pressure to be 70 pounds per square inch. Applying formula **144,** the M. E. P. $= .9\,[.937\,(70 + 14.7) - 17] = 56.13$ pounds per square inch. Letting $d =$ diameter of cylinder,

$$\text{I. H. P.} = \frac{d^2 \times .7854 \times 56.13 \times 500}{33,000} = 97.5,\ \text{or}$$

$$d = \sqrt{\frac{97.5 \times 33,000}{.7854 \times 56.13 \times 500}} = 12.08 \text{ inches, or say 12 inches.}$$

Taking the ratio of stroke to diameter of cylinder as $1\frac{1}{2}$, we have stroke $= 12 \times 1\frac{1}{2} = 18$ inches. The number of revolutions of the crank is

$$\frac{500 \times 6}{18} = 166\frac{2}{3} \text{ revolutions per minute.}$$

(1267) A combination of two single-cylinder engines of exactly the same description and dimensions, which have their cranks rigidly connected to a common crank-shaft and take the steam at the same pressure, is called a *duplex* engine.

Compound engines are those having two cylinders, of which the working lengths are usually the same, but the diameter of one, the high-pressure cylinder, is less than that of the other, the low-pressure cylinder, and the steam, instead of entering both cylinders at boiler-pressure, enters first the high-pressure cylinder, and is exhausted from there into the low-pressure cylinder

(1268) One in which the cylinder is in a vertical or upright position.

(1269) The stroke of an engine is the distance passed over by the piston when moving from one end of the cylinder to the other end, and is equal to the *throw of the crank*, or to the diameter of the circle described by the center of the crank-pin.

(1270) An eccentric is a disk, or wheel, so arranged upon a shaft that the center of the wheel and that of the shaft do not coincide. It is equivalent to a crank having

the same throw, and is used to give motion to the slide-valve.

(1271) It is the period during which the steam remaining in the cylinder after the exhaust-valve has closed is compressed as the piston continues the return stroke. It begins at the instant that the valve closes the port to the exhaust-steam.

(1272) It shortens the period of release and lengthens both the period of expansion and compression.

(1273) It permits an earlier cut-off, together with a greater range and more perfect steam distribution.

(1274) Using formula **143**,

$$\text{I. H. P.} = \frac{62.4 \times .7854 \times 18^2 \times \frac{24}{12} \times 2 \times 175}{33,000} =$$

336.825 I. H. P.   Ans.

(1275) By setting the cranks at right angles, both engines can not be on a dead-center at the same time.

(1276)   See Arts. **2097** and **2098.**

(1277)   By the *bore* of a cylinder is meant its diameter.

(1278)   Steam is called live steam when it leaves the boiler and before doing any work in the cylinder.   The energy stored in the live steam is potential energy.

(1279)   The fly-wheel supplies the force necessary to overcome the retarding effect of compression.

(1280)   (*a*) The dead-center positions occur when the piston reaches the end of its stroke, and the centers of the cross-head pin, crank-pin, and crank-shaft are in the same straight line.

(*b*) Twice.   .

(1281)   A steam-engine indicator is an instrument which draws a diagram showing the pressure of the steam

in the cylinder at every point of the strokc.   See Fig. 679 for method of fastening to cylinder.

(1282)   See Art. **2097.**

(1283)   See Art. **2039.**

(1284)   It is the resistance against being pushed into the condenser or the atmosphere which the exhaust-steam exerts on the piston.

(1285)   By period of release is meant that period during which the steam is escaping into the atmosphere or condenser.   The point of compression marks the end of release.

(1286)   See Art. **2049.**

(1287)   Two.   One spring is to resist any upward motion of the indicator-piston, and the other is to carry the drum back to its first position when the pull on the cord is discontinued.

(1288)   The back-pressure line would then fall below its present position a distance represented by a pressure of $\frac{5}{6} \times 14.7 = 12\frac{1}{4}$ pounds $= \frac{12\frac{1}{4}}{45} = .27$ inch, nearly.   Then, for the same mean effective pressure, the cut-off would be earlier.

(1289)   See Art. **2091.**

(1290)   See Art. **2039.**

(1291)   See Art. **2042.**

(1292)   Release is taking place.

(1293) · The varying pressures of the steam while being compressed.

(1294)   See Art. **2050.**

(1295)   See Art. **2055.**

(1296)   See Art. **2055.**

(1297)   Sec Art. **2078.**

(1298)   See Art. **2039.**

(1299) See Art. **2050.**

(1300) At the end. See Art. **2045.**

(1301) See Arts. **2048** and **2050.**

(1302) See Art. **2055.**

(1303) Using formula **144** and the constants in Tabl 44, the M. E. P. for $\frac{3}{10}$ cut-off is

$.9\,[.708\,(75 + 14.7) - 17] = 41.86$ lb. per sq. in. Ans.

For cut-off at $\frac{1}{2}$ stroke,

M. E. P. $= .9\,[.847\,(75 + 14.7) - 17] = 53.16$ lb. per sq. in. Ans

(1304) See Art. **2080.**

(1305) See Art. **2092.**

(1306) See Art. **2039.**

(1307) See Art. **2044.**

(1308) Closed. See Art. **2045** and Fig. 670 **(a)**.

(1309) See Art. **2045.**

(1310) Using formula **145,**

$$S = \frac{lR}{6} \text{ or } l = \frac{6\,S}{R} = \frac{350 \times 6}{175} = 12 \text{ inches.}\quad \textbf{Ans.}$$

(1311) See Art. **2080.**

(1312) See Arts. **2094** and **2098.**

(1313) See Art. **2039.**

(1314) See Art. **2043.**

(1315) See Art. **2052.**

(1316) Applying the rule, Art. **2056,**

$$\text{length} = \frac{96 \times 3}{12} = 24 \text{ inches.}\quad \textbf{Ans.}$$

(1317) Applying formula **145,**

$$S = \frac{lR}{6} = \frac{48 \times 50}{6} = 400 \text{ feet per minute.}\quad \textbf{Ans.}$$

(1318) See Art. **2085.**

(1319) See Arts. **2096** and **2097.**

(1320) See Art. **2044.**

(1321) See Art. **2044.**

(1322) Formula **143** gives

$$\text{I. H. P.} = \frac{PLAN}{33,000} = \frac{43.4 \times 1\frac{3}{4} \times 22^2 \times .7854 \times 2 \times 200}{33,000} =$$

300 H. P., nearly.   Ans.

(1323) See Art. **2061.**

(1324) Applying formula **145,**

$$S = \frac{lR}{6}, \text{ or } R = \frac{6S}{l};$$

therefore,

$$R = \frac{6 \times 750}{60} = 75 \text{ rev. per min.}   \textbf{Ans.}$$

(1325) See Art. **2081.**

(1326) See Art. **2095.**

AIR AND AIR COMPRESSION.

(1327) See Art. **2101.**

(1328) (*a*) 4 ft. = 48 in. A cubic inch of mercury weighs .49 lb. ; hence, the pressure exerted by 48 inches of mercury = $48 \times .49 = 23.52$ lb. per sq. in. Ans.

(*b*) Since the pressure of 1 atmosphere is 14.7 lb. per sq. in., a pressure of 23.52 lb. per sq. in. is equivalent to $23.52 \div 14.7 = 1.6$ atmospheres. Ans.

(1329) A pressure of 1 atmosphere will support a column of water 34 ft. high. Since the column of water is 15 ft. high, the height of the confined air is $34 - 15 = 19$ ft., or, in other words, the tension of the confined air in pounds per square inch is equal to the weight of a column of water 1 in. square and 19 ft. high. The pressure exerted by a column of water 1 ft. high and having a cross-section of 1 sq. in. $= 12 \times .03617 = .434$ lb. Hence the tension of the confined air $= .434 \times 19 = 8.246$ lb. per sq. in. Ans.

(1330) See Arts. **2117** and **2118.**

(1331) See Art. **2118.**

(1332) There would be no loss, because the air would have no opportunity to lose heat by radiation in the pipes. The heat stored in the air during compression would be available for useful work.

(1333) See Arts. **2121** and **2126.**

(1334) See Art. **2131.**

(1335) See Art. **2136.**

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(1336) See Art. **2145.**

(1337) (*a*) See Art. **2113.** (*b*) More work is required to compress air adiabatically.

(1338) Applying formula **157** and substituting,

$$p_1 = p\left(\frac{459 + t_1}{459 + t}\right) = 40 \times \left(\frac{459 + 55}{459 + 120}\right) =$$

$$40 \times \frac{514}{579} = \frac{20,560}{579} = 35.51 \text{ lb. per sq. in.} \text{Ans.}$$

(1339) (*a*) $14.7 + 9 = 23.7$ lb. per sq. in., the tension. Since the area of the piston remains constant, the volume at any point of the stroke will be proportional to the distance passed over by the piston; hence, we may substitute the latter for the former in formula **151.**

$$v_1 = \frac{p\,v}{p_1} = \frac{14.7 \times 80}{23.7} = 49.62 \text{ in., distance between piston}$$

and end of stroke. The distance passed over by piston $= 80 - 49.62 = 30.38$ in. Ans.

(*b*) Area of piston $= 80^2 \times .7854$ sq. in.
Volume of air at point of discharge $= 80^2 \times .7854 \times 49.62 = 249,417.91$ cu. in.

$$249,417.91 \div 1,728 = 144.34 \text{ cu. ft.} \text{Ans.}$$

(1340) Since the required horsepower is 25 and the loss is 35%, the horsepower of the engine must be $25 \div (100\% - 35\%) = 25 \div .65 = 38.46$ H. P.

To calculate the M. E. P., formula **144** may be used.

M. E. P. $= .9\,[.904\,(92 + 14.7) - 17] = 71:5$ lb. per sq. in.

To find the diameter of the steam-cylinder, substitute in formula **148.**

$$D = 79.6 \sqrt[3]{\frac{38.46}{1\frac{3}{8} \times 71.5 \times 340}} = 8.34; \text{ or say } 8\frac{3}{8} \text{ in.}$$

Length of stroke $= 8\frac{3}{8} \times 1\frac{3}{8} = 11\frac{1}{2}$ in. Consequently, the steam-cylinder should be $8\frac{3}{8} \times 11\frac{1}{2}$ in. Ans.

(1341) The steam-cylinder will show the greater I. H. P. The difference represents the horsepower required to overcome the friction of the moving parts of the compressor.

(1342) See Art. **2149.**

(1343) (*a*) Volume of cylinder $= \dfrac{20^3 \times .7854 \times 32}{1,728} =$ 5 8178 cu. ft.

$32 - 26 = 6$ in., length of stroke unfinished.

The volume at discharge is $\frac{6}{32}$ of volume at beginning of stroke, or $\frac{6}{32} \times 5.8178 = 1.0908$ cu. ft.   Ans.

(*b*) To calculate the weight, substitute in formula **161**, taking the values of P, V, and T at beginning of stroke.

$$W = \frac{PV}{.37052\,T} = \frac{14.7 \times 5.8178}{.37052 \times (459 + 76)} = .43143 \text{ lb.}\quad \text{Ans.}$$

(1344) Since the area of the cylinder remains constant, any variation in the volume will be proportional to the distance between the piston and end of stroke; hence, we may substitute the latter for the volume in formula **150.**

$$p_1 = \frac{p\,v}{v_1} = \frac{14.7 \times 32}{6} = 78.4 \text{ lb. per sq. in.}\quad \text{Ans.}$$

(1345) Using formula **150,**

$$p_1 = \frac{p\,v}{v_1} = \frac{(3 \times 14.7) \times 1}{2.5} = 17.64 \text{ lb.}\quad \text{Ans.}$$

(1346) Applying formula **159** and substituting,

$$V = \frac{.37052\,W\,T}{P} = \frac{.37052 \times 7.14 \times (459 + 75)}{1.5 \times 14.7} =$$
64.068 cu. ft.   Ans.

(1347) $p = 3\frac{1}{2}$ atmospheres $= 3\frac{1}{2} \times 14.7 = 51.45$ lb. per sq. in.

Applying formula **154** and substituting,

$$p : W :: p_1 : W_1.$$
$$51.45 : 13 :: p_1 : 2.$$

$$p_1 = \frac{2 \times 51.45}{13} = 7.915 \text{ lb. per sq. in.}\quad \text{Ans.}$$

(**1348**) Volume at beginning of stroke =

$$\frac{48^2 \times .7854 \times 60}{1,728} = 62.832 \text{ cu. ft.}$$

Substituting in formula **161** to obtain the weight of the air,

$$W = \frac{PV}{.37052\ T} = \frac{14.7 \times 62.832}{.37052 \times (459 + 60)} = 4.8031 \text{ lb.}$$

Volume at time of discharge =

$$\frac{48^2 \times .7854 \times (60 - 50)}{1,728} = 10.472 \text{ cu. ft.}$$

To calculate the tension, substitute in formula **158**, taking the values of T and V at the time of discharge and the value of W as 4.8031.

$$P = \frac{.37052\ W\ T}{V} = \frac{.37052 \times 4.8031 \times (459 + 130)}{10.472} =$$

100.096 lb. per sq. in. Ans.

(**1349**) Applying formula **159** and substituting,

$$V = \frac{.37052\ W\ T}{P} = \frac{.37052 \times 1 \times (459 + 127)}{27} = 8.042 \text{ cu. ft.}$$

Ans.

(**1350**) A pressure of 4,000 lb. per sq. ft. is equivalent to $\frac{4,000}{144}$, or 27.777 lb. per sq. in.   Using formula **159**,

$$V = \frac{.37052\ W\ T}{P} = \frac{.37052 \times .5 \times 559}{27.777} = 3.728 \text{ cu. ft.} \text{ Ans.}$$

(**1351**) Applying formula **156** and substituting,

$$v_1 = v\left(\frac{459 + t_1}{459 + t}\right) = 4 \times \left(\frac{459 + 115}{459 + 40}\right) = 4.6012 \text{ cu. ft.} \text{ Ans.}$$

(**1352**) See Art. **2147**.

(**1353**) Since the ordinary temperature is given in each case, we add 459° to obtain the corresponding absolute temperatures.

459° + 32° = 491°; 459° + 212° = 671°; 459° + 62° = 521°; 459° + 0° = 459°; 459° − 40° = 419°.

(1354) $P = 10$ atmospheres $= 10 \times 14.7 = 147$ lb. per sq. in. Applying formula **160** and substituting,

$$T = \frac{PV}{.37052\,W} = \frac{147 \times 4}{.37052 \times 3.5} = 453.417°.$$

$453.417° - 459° = -5.583°$, or $5.583°$ below zero.   Ans.

(1355) See Art. **2134.**

(1356) Applying formula **150** and substituting,

$$p_1 = \frac{p\,v}{v_1} = \frac{130 \times 11.798}{75} = 20.45 \text{ lb. per sq. in.} \text{ Ans.}$$

(1357) Applying formula **160**, $T = \dfrac{PV}{.37052\,W}$.

Substituting, $T = \dfrac{18 \times 14}{.37052 \times 1.2} = 566.77°$. $566.77° - 459° = 107.77°$. Ans.

(1358) Applying formula **156** and substituting,

$$v_1 = v \left(\frac{459 + t_1}{459 + t}\right) = 21 \times \left(\frac{459 + 420}{459 + 60}\right) = 35.57 \text{ cu. ft.} \text{ Ans.}$$

(1359) To obtain absolute pressure, 1 atmosphere must be added to the gauge-pressure. $6 + 1 = 7$ atmospheres. Substituting in formula **161,**

$$W = \frac{PV}{.37052\,T} = \frac{14.7 \times 7 \times 12}{.37052 \times (459 + 90)} = 6.07033 \text{ lb.},$$

weight of 12 cubic feet. $6.07033 \div 12 = .50586$ lb., weight per cubic foot.   Ans.

(1360) .5 lb. $= 8$ oz.   1 lb. 6 oz. $= 22$ oz.

Applying formula **154** and substituting,

$$p : W :: p_1 : W_1.$$
$$14.7 : 8 :: p_1 : 22.$$

$$p_1 = \frac{14.7 \times 22}{8} = 40.425 \text{ lb. per sq. in.} \textbf{Ans.}$$

(1361) Apply formula **156.** $v_i = v\left(\dfrac{459 + t_i}{459 + t}\right)$.
Substituting,

$$v_i = 4{,}516\left(\frac{459+80}{459+260}\right) = 4{,}516 \times \frac{539}{719} = 3{,}385.42 \text{ cu. in.}$$

$3{,}385.42 \div 1{,}728 = 1.96$ cu. ft.   Ans.

(1362) $P = 1\frac{1}{4} \times 14.7$ lb. per sq. in.
Applying formula **161** and substituting,

$$W = \frac{PV}{.37052\,T} = \frac{1\frac{1}{4} \times 14.7 \times 55}{.37052 \times (459 + 88)} = \frac{1{,}010.625}{202.67444} = 4.986 \text{ lb.}$$
Ans.

(1363) Since the temperature and volume in both vessels are the same, the pressure of the mixture will be equal to the sum of the pressures.

2 atmospheres $= 2 \times 14.7 = 29.4$ lb. per sq. in.

$29.4 + 40 = 69.4$ lb. per sq. in.   Ans.   See Art. **2167.**

(1364) We would understand that the mercury had fallen 7 inches, and that there was enough air in the condenser to produce a pressure of $\dfrac{30-23}{30} \times 14.7$, or $\frac{7}{30} \times 14.7 = 3.43$ lb. per sq. in.   Ans.   See Art. **2155.**

(1365) $144 \times 14.7 = 2{,}116.8$ lb. per sq. ft.   Ans.

(1366) If the weight of 3 cu. ft. under a pressure of 30 lb. per sq. in. is .27 lb., the weight per cu. ft. $= \dfrac{.27}{3} = .09$ lb.
Applying formula **154** and substituting,

$$p : W :: p_i : W_i, \text{ or } 30 : .09 :: 65 : W_i.$$

$$W_i = \frac{.09 \times 65}{30} = .195 \text{ lb.}   \text{Ans.}$$

(1367) To find the absolute temperature, we substitute in formula **160,** the values of P, V, and W given in Question 1366.

$$T = \frac{PV}{.37052\,W} = \frac{30 \times 3}{.37052 \times .27} = 899.64°.$$

Ordinary temperature $= 899.64° - 459° = 440.64°$.   **Ans.**

(**1368**) Since the pressures and volumes are unequal, we apply formula **162** in order to obtain the tension of the mixture.

$$P = \frac{p\,v + p_1\,v_1}{V}.$$

Substituting,

$$P = \frac{14.7 \times 12 + 3 \times 14.7 \times 8}{20} = \frac{176.4 + 352.8}{20} =$$

26.46 lb. per sq. in. Ans.

(**1369**) Applying formula **163** and substituting,

$$V = \frac{p\,v + p_1\,v_1}{P} = \frac{14.7 \times 12 + 3 \times 14.7 \times 8}{24} = 22.05 \text{ cu. ft.}$$
Ans.

(**1370**) See Art. **2155.**

(**1371**) Since a cubic inch of mercury weighs .49 lb., $\frac{1}{40}$ of a cubic inch weighs $\frac{1}{40} \times .49 = \frac{.49}{40}$ lb. Consequently, a height of $\frac{1}{40}$ in. of mercury is equivalent to a pressure of $\frac{.49}{40}$ lb. per sq. in. 1 sq. ft. = 144 sq. in. The equivalent pressure upon a sq. ft. $= \frac{.49}{40} \times 144 = 1.764$ lb. Ans.

(**1372**) (*a*) See Art. **2155.**

(*b*) The height of the mercury in the tube shows the number of inches of vacuum.

Since the mercury column is $4\frac{1}{2}$ inches shorter than the barometer column, the gauge will show $30 - 4\frac{1}{2} = 25\frac{1}{2}$ inches vacuum. Ans.

(**1373**) Since a column of mercury 30 in. high will support a column of water 34 ft., 1 in. of mercury will support a column of water of $\frac{1}{30} \times 34$, or $\frac{34}{30}$ ft. in height.

Hence, 27 inches of mercury will support $27 \times \frac{34}{30} = 30.6$ ft. of water. Ans.

(**1374**) See Art. **2142.**

(**1375**) See Art. **2140.** Each rock-drill requires a receiver volume of 10 cubic feet; therefore, to supply 8 rock-drills, the volume of the receiver should be $8 \times 10 = 80$ cu. ft. Ans.

(**1376**) See Art. **2119.** (1) There is a loss due to useless heating of the air during the compression; this is reduced by the water-jacket or by water injection. (2) The loss due to the friction of the engine and compressor can only be reduced by careful workmanship and design in the building of the compressor. (3) There is a slight loss due to leakage and friction of air in pipes. The loss due to friction becomes large when the pipe is very long and of small diameter; therefore, this loss is reduced to a minimum by using large pipes for conveying the air.

HYDROMECHANICS AND PUMPING.

(1377) (*b*) To obtain the discharge in cubic feet **per** second, apply formula **180.**

$$Q_a = .41\, b\, \sqrt{2\,g}\,[\sqrt{h^3} - \sqrt{h_1{}^3}] =$$
$$.41 \times \tfrac{2\,5}{1\,2} \times \sqrt{2 \times 32.16}\,[\sqrt{(5\tfrac{1}{2})^3} - \sqrt{(3\tfrac{1}{2})^3}] = 52.21 \text{ cu. ft. per sec.}$$
Ans.

(*a*) Area of weir $= b\,d = 2.5 \times 2 = 5$ sq. ft.
Apply formula **179.**

$$v_m = \frac{Q_a}{b\,d} = \frac{52.21}{5} = 10.44 \text{ ft. per sec.} \quad \text{Ans.}$$

(*c*) To get the discharge in gallons per hour, multiply (*b*) by 60×60 (seconds in an hour) and by 7.48 (gallons in a cubic foot). Thus, $52.21 \times 60 \times 60 \times 7.48 = 1,405,910.9$ gal. per hour. Ans.

(1378) First find the coefficient of friction by using formula **182** and Table 45.

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l}} = 2.315 \sqrt{\frac{76 \times 7.5}{.025 \times 12,000}} = 3.19 \text{ ft. per sec.}$$

From the table, $f = .0243$ for $v_m = 3$, and .023 for $v_m = 4$; the difference is $.0013 = $ difference for a difference of velocity of 1 ft. per sec. Then, $.0013 \times .19$ ft. per sec. $= .000247$, or say .0002, using but four decimal places $=$ difference for a difference of velocity of .19 ft. per sec. Therefore, $f = .0243 - .0002 = .0241$, or say .024.

Use formula **186.** Substitute in it the value of f here found, and multiply by 60 to get the discharge per minute.

$$Q = .09445\, d^2 \sqrt{\frac{h\,d}{f\,l + .125\,d}} \times 60 =$$

$$.09445 \times 7.5^2 \sqrt{\frac{76 \times 7.5}{.024 \times 12,000 + .125 \times 7.5}} \times 60 = 447.7 \text{ gal.}$$
per min. Ans.

(1379) (a) Use formula **181.**

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l + .125\,d}} \times 60 =$$

$$2.315 \sqrt{\frac{76 \times 7.5}{.024 \times 12,000 + .125 \times 7.5}} \times 60 = 195.08 \text{ ft. per min.}$$
Ans.

(b) 447.7 gal. per min. ÷ 60 = 7.46⅛ gal. per sec. = 1 cu. ft. per sec., nearly. **Ans.**

(1380) Use formula **167.**

$$v = \sqrt{2\,g\,h} = \sqrt{2 \times 32.16 \times 10} = 25.36 \text{ ft. per sec.}   \textbf{Ans.}$$

(1381) Use formulas **185** and **183.**

$$v_m = \frac{24.51\, Q}{d^2} = \frac{24.51 \times 42,000}{6.5^2 \times 60 \times 60} = 6.768 \text{ ft. per sec.}$$

From Table 45, $f = .021$ for $v_m = 6.768$; hence,

$$h = \frac{f\,l\,v_m^{\,2}}{5.36\,d} + .0233\, v_m^{\,2} =$$

$$\frac{.021 \times 1,500 \times 6.768^2}{5.36 \times 6.5} + .0233 \times 6.768^2 = 42.48 \text{ ft.}   \textbf{Ans.}$$

(1382) Area of top or bottom of cylinder equals 20² × .7854 = 314.16 sq. in. Area of cross-section of pipe = (⅜)² × .7854 = .1104 sq. in.   25 lb. 10 oz. = 25.625 lb.   25.625 ÷ .1104 = 232.11 lb. pressure per sq. in. on top or bottom exerted by the weight and piston.

Pressure due to a head of 10 ft. = .434 × 10 = 4.34 lb. per sq. in.

Pressure due to a head of 13 ft. = .434 × 13 = 5.642 lb. per sq. in.

(Since a column of water 1 ft. high **exerts a pressure of** .434 lb. per sq. in.   See Art. **2289.)**

(*a*) Pressure on bottom = pressure due to weight + pressure due to head of 13 ft. = 232.11 + 5.64 = 237.75 lb. per sq. in. Ans.

(*b*) Pressure on the top = pressure due to weight + pressure due to head of 10 feet = 232.11 + 4.34 = 236.45 lb. per sq. in. Ans.

(*c*) Total pressure, or equivalent weight on the bottom, = 237.752 × 314.16 = 74,692.17 lb.   Ans.

(**1383**)   .434 × 1½ = .651 lb., pressure due to the head of water in the cylinder above the orifice.

236.45, pressure on top per sq. in. + .651 = 237.1, total pressure per sq. in.   Area of orifice = 1² × .7854 = .7854 sq. in.

$$.7854 \times 237.1 = 186.22 \text{ lb.} \quad \textbf{Ans.}$$

(**1384**)   First find the coefficient of friction by formula **182** and Table 45.

$$v_m = 2.315 \sqrt{\frac{h\,d}{.025\,l}} = 2.315 \sqrt{\frac{120 \times 4}{.025 \times 4,000}} = 5.072 \text{ ft. per sec.,}$$

or say 5 ft. per sec.

From the table, $f = .023$ for $v_m = 4$ and $.0214$ for $v_m = 6$.

$$\frac{.023 - .0214}{6 - 4} = .0008. \quad .023 - .0008 = .0222 = f \text{ for } v_m = 5$$

Use formula **182**, because the pipe is longer than 10,000 times its diameter.

Hence, $v_m = 2.315 \sqrt{\dfrac{120 \times 4}{.0222 \times 4,000}} = 5.38$ ft. per sec. Ans.

(**1385**)   Use formulas **182** and **181**.

$$v_m = 2.315 \sqrt{\frac{120 \times 4}{.025 \times 2,000}} = 7.17 \text{ ft. per sec.}$$

From the table, $f = .0214$ for $v_m = 6$ and $.0205$ for $v_m = 8$.

$$\frac{.0214 - .0205}{8 - 6} = .00045 \text{ decrease for an increase of velocity}$$

of 1 ft. per sec.   7.17 − 6 = 1.17.   .00045 × 1.17 = .0005265, total decrease. $f = .0214 - .0005265 = .0208735$, or .0209, using four figures.

Hence, the velocity of discharge =

$$v_m = 2.315 \sqrt{\frac{120 \times 4}{.0209 \times 2,000 + .125 \times 4}} = 7.79 \text{ ft. per sec.}$$
Ans.

(1386) (a) $f = .0205$ for $v_m = 8$. Therefore, using formula 183,

$$h = \frac{.0205 \times 5,280 \times 8^2}{5.36 \times 10} + .0233 \times 8^2 = 130.73 \text{ ft.}\quad \text{Ans.}$$

(b) Using formula 184, $Q = .0408\, d^2\, v_m = .0408 \times 10^2 \times 8 = 32.64$ gal. per sec. $32.64 \times 60 \times 60 = 117,504$ gallons per hour. Ans.

(1387) A column of water 1 inch square and 2.304 ft. high weighs 1 lb. ; hence, to produce a pressure of 30 lb. per sq. in., it will require a column of water $2.304 \times 30 = 69.12$ ft. high = head. Using formula 172,

$$v = .98\sqrt{2gh} = .98\sqrt{2 \times 32.16 \times 69.12} = 65.34 \text{ ft. per sec. Ans.}$$

(1388) (a) 36 in. = 3 ft. A column of water 1 inch square and 1 ft. long weighs .43403 lb. $.43403 \times 43 = 18.6633$ lb. per sq. in. on the bottom of the cylinder. $.43403 \times 40 = 17.3612$ lb. per sq. in. on the top of the cylinder. Area of base of cylinder = $20^2 \times .7854 = 314.16$. $314.16 \times 18.6633 = 5,863.26$ lb., the total pressure on the bottom. Ans.

(b) $314.16 \times 17.3612 = 5,454.19$ lb., total pressure on top. Ans.

(1389) Use formula 168. $h = \dfrac{33^2}{64.32} = 16.931$ ft. per sec. Ans.

(1390) (a) Use formula 178 and multiply by $7.48 \times 60 \times 60$ to reduce cubic feet per second to gallons per hour.

$$Q_a = .41 \times \tfrac{24}{12} \times \sqrt{2 \times 32.16 \times \left(1\tfrac{8}{12}\right)^2} \times 7.48 \times 60 \times 60 =$$
$$216,551 \text{ gal. per hr. Ans.}$$

(b) By formula 179,

$$v_m = \frac{Q_a}{b\,d} = \frac{.615 \times \tfrac{2}{3} \times \tfrac{24}{12} \times \sqrt{2 \times 32.16 \times \left(1\tfrac{8}{12}\right)^2}}{\tfrac{24}{12} \times 1\tfrac{8}{12}} =$$
$$3.676 \text{ ft. per sec. Ans.}$$

(1391) $f = .0193$ for $v_m = 12$.   Therefore, using formula **183**,

$$h = \frac{f l v_m{}^2}{5.36 d} + .0233 v_m{}^2 = \frac{.0193 \times 6,000 \times 12^2}{5.36 \times 3} + (.0233 \times 12^2) =$$
1,040.37 ft.   Ans.

(1392) $\dfrac{5^2 \times .7854}{144} =$ area of pipe in sq. ft.   Using formula **165**,

$$Q = A v = \frac{5^2 \times .7854}{144} \times 7.2 = \text{discharge in cu. ft. per sec.}$$

$\dfrac{5^2 \times .7854}{144} \times 7.2 \times 7.48 \times 60 \times 60 \times 24 = 634,478$ gal. discharged in 1 day.   Ans.

(1393) 38,000 gal. per hour $= \dfrac{38,000}{60 \times 60}$ gal. per sec. $= Q$. Using formula **185**,

$$v_m = \frac{24.51 \, Q}{d^2} = \frac{24.51 \times 38,000}{5.5^2 \times 60 \times 60} = 8.5526 \text{ ft. per sec.}\quad \text{Ans.}$$

(1394)   Use formula **178.**

(*a*) $Q_a = .41 \times b\sqrt{2 g d^3} = .41 \times \frac{23}{12} \times \sqrt{2 \times 32.16 \times (\frac{36}{12})^3} =$
38.44 ft. per sec.   Ans.

(*b*) $Q = \dfrac{Q_a}{.615} = \dfrac{38.44}{.615} = 62.5$ cu. ft. per sec.   **Ans.**

(1395)   Use formulas **167** and **169.**

(*a*) $v = \sqrt{2 g h} = \sqrt{2 \times 32.16 \times 45} = 53.8$ ft. per sec.   Ans.

(*b*) $2.304 \times 10 = 23$ ft., nearly = height of a column of water which will give a pressure of 10 lb. per sq. in. $45 + 23 = 68$ ft.

$$v = \sqrt{2 g (h_1 + h)} = \sqrt{2 \times 32.16 \times 68} = 66.153 \text{ ft. per sec.}$$
Ans.

(1396)   Use formula **184.**

$Q = .0408 \, d^2 \, v_m = .0408 \times 6^2 \times 7.5 = 11.016$ gal. per sec.   Ans.

(1397) Head $= 41 \div .434 = 94.47$ ft.
Using formula **172,**

$$v = .98 \sqrt{2\,g\,h} = .98\sqrt{2 \times 32.16 \times 94.47} = 76.39 \text{ ft. per sec.}$$
Ans.

(1398) Divide by 60×60 to get the discharge in gal. per sec., and by 7.48 to obtain the discharge in cu. ft. per sec.

$$\text{Area in sq. ft.} = \frac{4^2 \times .7854}{144} = .087267.$$

$$v_m = \frac{Q}{A} = \frac{12,000}{60 \times 60 \times 7.48 \times .087267} = 5.106 \text{ ft. per sec. Ans.}$$

(1399) (a) Area of pump-piston $= (\tfrac{1}{2})^2 \times .7854 = .19635$ sq. in.
Area of plunger $= 10^2 \times .7854 = 78.54$ sq. in.

Pressure per sq. in. exerted by piston $= \dfrac{100}{.19635}$ pounds.

Hence, according to Pascal's law, the pressure on the plunger is $\dfrac{100}{.19635} \times 78.54 = 40,000$ lb. Ans.

(b) According to the principle given in Art. **2181,**
$D \times 1\tfrac{1}{2}$ inches $= W \times$ distance moved by plunger, or $100 \times 1\tfrac{1}{2} = 40,000 \times$ required distance; hence, the required distance $= \dfrac{100 \times 1\tfrac{1}{2}}{40,000} = .00375$ in. Ans.

(1400) (a) Use formula **180,** and multiply by 7.48 and 60 to reduce the discharge from cu. ft. per sec. to gal. per min.

$$Q_a = .41\,b\,\sqrt{2\,g}\left[\sqrt{h^3} - \sqrt{h_1^3}\right] \times 60 \times 7.48 =$$

$$.41 \times 1\tfrac{1}{2} \times \sqrt{64.32}\left[\sqrt{(9 + \tfrac{9}{12})^3} - \sqrt{9^3}\right] \times 60 \times 7.48 =$$
$$13,491.22 \text{ gallons per minute. Ans.}$$

(b) In the second case,

$$Q_a = .41 \times \tfrac{20}{12} \times \sqrt{64.32}\left[\sqrt{(9 + \tfrac{11}{12})^3} - \sqrt{9^3}\right] \times 60 \times 7.48 =$$
$$13,322.47 \text{ gallons per minute. Ans.}$$

(**1401**)   Area of weir $= 14 \times 20 \div 144$ sq. ft.   Use formula **166,** and divide by 60×7.48 to reduce gal. per min. to cu. ft. per sec.

(*a*) $v_m = \dfrac{Q}{A} = \dfrac{13{,}491.22 \times 144}{60 \times 7.48 \times 14 \times 20} = 15.46$ ft. per sec. Ans.

(*b*) $\dfrac{13{,}322.47 \times 144}{60 \times 7.48 \times 14 \times 20} = 15.264$ ft. per sec.   Ans.

(**1402**)   In Art. **2197** it is stated that the theoretical mean velocity is $\frac{2}{3}\sqrt{2gh}$.   Hence,

$$v_m = \tfrac{2}{3}\sqrt{2 \times 32.16 \times 3} = 9.26 \text{ ft. per sec.   Ans.}$$

(**1403**)   (*a*) 4 ft. 9 in. $= 4.75$ ft. $19 - 4.75 = 14.25$. Using formula **170,**

$$R = \sqrt{4hy} = \sqrt{4 \times 4.75 \times 14.25} = 16.454 \text{ ft.   Ans.}$$

(*b*) $19 - 4.75 = 14.25$ ft.   Ans.

(*c*) $19 \div 2 = 9.5$.   Greatest range $= \sqrt{4 \times 9.5^2} = 19$ ft. Ans.

(**1404**)   Use formulas **182** and **186.**

$$v_m = 2.315\sqrt{\dfrac{hd}{fl}} = 2.315\sqrt{\dfrac{25 \times 5}{.025 \times 1{,}300}} = 4.54 \text{ ft. per sec.}$$

From the table, $f = .023$ for $v_m = 4$, and $.0214$ for $v_m = 6$. $\dfrac{.023 - .0214}{2} \times .54 = .000432$. $f = .023 - .000432 = .022568$.

$$Q = 60 \times 60 \times .09445 \times 5^2 \times \sqrt{\dfrac{25 \times 5}{.022568 \times 1{,}300 + .125 \times 5}} =$$
17,368.95 gal. per hr.   Ans.

(**1405**)   Obtain the values by approximating to those given in Table 45.   Thus, for $v_m = 2$, $f = .0265$, and for $v_m = 3$, $f = .0243$.

Difference $= .0022$. $.0022 \times .37 = .000814$, or say $.0008$. $.0265 - .0008 = .0257 = f$ for $v_m = 2.37$.   Ans.

$.0243 - .023 = .0013$. $.0013 \times .19 = .000247$, or say $.0002$. $.0243 - .0002 = .0241 = f$ for $v_m = 3.19$.   Ans.

$.023 - .0214 = .0016.$ $\dfrac{.0016}{2} \times 1.8 = .00144$, or say $.0014$.

$.023 - .0014 = .0216 = f$ for $v_m = 5.8$. Ans.

$.0214 - .0205 = .0009.$ $\dfrac{.0009}{2} \times 1.4 = .00063$, or say $.0006$.

$.0214 - .0006 = .0208 = f$ for $v_m = 7.4$.

$.0205 - .0193 = .0012.$ $\dfrac{.0012}{4} \times 1.83 = .000549$, or say

$.0005.$ $.0205 - .0005 = .02 = f$ for $v_m = 9.83$. Ans.

$.0205 - .0193 = .0012.$ $\dfrac{.0012}{4} \times 3.5 = .00105$, or say $.0011$.

$.0205 - .0011 = .0194 = f$ for $v_m = 11.5$. Ans.

(1406) The specific gravity of sea-water is 1.026 (see table of Specific Gravity); hence, the weight of a cubic foot of sea-water $= 1.026 \times 62.5 = 64.1$ lb.

Total area of cube $= \dfrac{10.5^2 \times 6}{144}$ sq. ft. 1 mile $= 5,280$ ft.

Hence, total pressure on cube $= \dfrac{10.5^2 \times 6}{144} \times 5,280 \times 3.5 \times 64.1 = 5,441,609.25$ lb. Ans.

(1407) (*b*) The pressure per square inch equals the weight of a volume of water 1 in. square and 12 in. high; that is, it equals

$1 \times 1 \times .03617 \times 12 = .434$ lb., nearly. Ans.

(*a*) Total pressure on the bottom = area of bottom in square inches multiplied by the pressure per square inch = $8^2 \times .7854 \times .434 = 21.82$ lb. Ans.

(1408) 8,000 gal. per hr. $= \dfrac{8,000}{60}$, or $133\frac{1}{3}$ gal. per min.

Plunger speed per min. $= 7 \times 10 = 70$ ft. Applying formula **190** and substituting,

$$d = 5.535 \sqrt{\dfrac{G}{S}} = 5.535 \sqrt{\dfrac{133\frac{1}{3}}{70}} = 7\frac{5}{8}\text{ in.}\quad\text{Ans.}$$

(1409) See Art. **2266.**

(1410) The height to which a pump will lift water is directly proportional to the atmospheric pressure; that is, proportional to the length of the mercury column.

Letting x represent the height to which the pump will lift the water on top of the mountain, we have the proportion, $30 : 22 :: 25.5 : x$, or $30\,x = 22 \times 25.5$; whence, $x = 18.7$ ft.   Ans.

(1411)   Area of dam $= 40 \times 12 = 480$ sq. ft.
$\frac{1}{2} \times 12 = 6$ ft., depth of center of gravity below the level of the liquid.
The total pressure on the dam $= 40 \times 12 \times 6 \times 62\frac{1}{2} = 180{,}000$ lb.   Ans.

(1412)   (*a*) Apply formula **190.**

$$d = 5.535 \sqrt{\tfrac{150}{100}} = 15 \text{ inches.}\quad\text{Ans.}$$

(*b*) Use formula **195.**

$$d_1 = .35 \sqrt{G} = .35 \sqrt{750} = 10 \text{ inches.}\quad\text{Ans.}$$

(*c*) Use formula **196.**

$$d_1 = .25 \sqrt{G} = .25 \sqrt{750} = 7 \text{ inches.}\quad\text{Ans.}$$

(1413)   First obtain the coefficient of friction from formula **182** and Table 45.

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l}} = 2.315 \sqrt{\frac{40 \times 6}{.025 \times 840}} = 7.82 \text{ ft. per sec.}$$

From the table, $f = .0214$ for $v_m = 6$ and $.0205$ for $v_m = 8$.
$\dfrac{.0214 - .0205}{2} = .00045$ decrease for an increase of velocity of 1 ft. per sec. $7.82 - 6 = 1.82$.
$.00045 \times 1.82 = .0008$, nearly, total decrease. $f = .0214 - .0008 = .0206$.
To obtain the discharge in gal. per sec., substitute this value of f in formula **186,** and multiply by 60×60 to get the discharge in gal. per hr.

$$Q = .09445\,d^2 \sqrt{\frac{h\,d}{f\,l + .125\,d}} \times 60 \times 60 =$$

$$.09445 \times 6^2 \times \sqrt{\frac{40 \times 6}{.0206 \times 840 + .125 \times 6}} \times 60 \times 60 =$$
$$44{,}553.6 \text{ gal. per hr.}\quad\text{Ans.}$$

(1414) If the area of the tube is $\frac{1}{4}$ sq. in., and that of the cylinder 80 sq. in., a force of 80 lb. on the small piston will raise a weight of $\frac{80}{\frac{1}{4}} \times 80 = 12,800$ lb. on the large piston. Since the length between the hand and the fulcrum is $7\frac{1}{2}$ times the distance between the piston-rod and the fulcrum, a force of 80 lb. on the end of the lever will raise a weight of $7\frac{1}{2} \times 12,800 = 96,000$ lb. **Ans.**

(1415) (*a*) Using formula **190**,

$$d = 5.535 \sqrt{\frac{200}{2 \times 150}} = 4\frac{1}{2} \text{ in.} \quad \textbf{Ans.}$$

(*b*) Use formula **195**.

$$d_1 = .35 \sqrt{G} = .35 \sqrt{200} = 5 \text{ in.} \quad \textbf{Ans.}$$

(*c*) Use formula **196**.

$$d_1 = .25 \sqrt{G} = .25 \sqrt{200} = 3\frac{1}{2} \text{ in.} \quad \textbf{Ans.}$$

(*d*) Applying formula **192** and substituting,

$$H = .00038 \, G h = .00038 \times 200 \times 250 = 19 \text{ H. P.} \quad \textbf{Ans.}$$

(1416) (*a*) Since the pressure exerted by a column of water 1 foot high = .434 lb. per sq. in., the pressure exerted by a column of water 210 ft. high = 210 × .434 = 91.14 lb. per sq. in. Ans.

(*b*) Applying formula **167** and substituting,

$$v = \sqrt{2 g h} = \sqrt{2 \times 32.16 \times 210} = 116.22 \text{ ft. per sec.} \quad \text{Ans.}$$

(1417) To calculate the diameter of the steam-cylinder, we apply formula **194**. But we must first obtain the value of *H*, or the horsepower, by formula **192**. $H = .00038 \, G h$.

Substituting, $H = .00038 \times \dfrac{27,000}{60} \times 240 = 41.04$ H. P. for both sides of the pump. $\dfrac{41.04}{2} = 20.52$ H. P. for each side.

Substituting in formula **194**,

$$D = 205 \sqrt{\frac{20.52}{90 \times 85}} = 10\frac{3}{4} \text{ inches.} \quad \textbf{Ans.}$$

Apply formula **190.**

27,000 gal. per hr. $= \dfrac{27,000}{60}$, or 450 gal. per min. for **both**

sides. $\dfrac{450}{2} = 225$ gal. for one side $= G$.

$$d = 5.535 \sqrt{\dfrac{G}{S}} = 5.535 \sqrt{\dfrac{225}{90}} = 8\tfrac{3}{4} \text{ inches.}   \textbf{Ans.}$$

(1418)   (*a*) A column of water 1 foot high and having a cross-section of 1 sq. in. weighs .434 lb.   Hence, the pressure per sq. in. at the bottom of the stand-pipe $= .434 \times 70 = 30.38$ lb. per sq. in.   Ans.

(*b*) At a distance of 30 ft. from the top of the water **the** pressure is $.434 \times 30 = 13.02$ lb. per sq. in.   Ans.

(1419) ' See Art. **2216.**

(1420)   (*a*) Apply formula **191.**

$G = .03264\, d^{2} S = .03264 \times 14^{2} \times 100 = 639.744$ gal. **per** min. due to one side of pump. $639.744 \times 2 = 1,279.488$ gal., total discharge per minute. $1,279.488 \times 60 = 76,769.28$ gal. per hour.   Ans.

(*b*) To obtain the height to which water can be raised, we apply formula **193;** but, before we can substitute in this formula, we must obtain the horsepower by applying the formula $H = \dfrac{P L A N}{33,000}$.   Remembering that $L \times N =$ piston speed, we have

$$H = \dfrac{45 \times 22^{2} \times .7854 \times 100}{33,000} = 51.8364 \text{ H. P.}$$

Substituting in formula **193,**

$$h = \dfrac{H}{.00038\, G} = \dfrac{51.8364}{.00038 \times 639.744} = 213.22 \text{ feet.}   \textbf{Ans.}$$

(1421) $307 \times .434 = 133.238$ lb. per sq. in.   Ans.

(1422)   The time of making the stroke depends simply on the acceleration of the pit-work, which in turn depends

solely on the difference between the weight of the pit-work
and water column minus the frictional resistances.   Now, if
this difference is too great, the stroke will be made too
quickly for safety and convenience, and, to obviate this, the
weight of the descending pit-work must be made less or the
weight of the ascending water column greater.   This is ac-
complished by balancing the pit-work, as explained in Arts.
2247 to **2249.**

(**1423**)   First find the value of f from Table 45.

$f = .0243$ for $v_m = 3$ and .023 for $v_m = 4.$

Difference $= .0013.$ $3.3 - 3 = .3.$ Then, $.0013 \times .3 =$
$.00039$, total decrease. $f = .0243 - .00039 = .02391.$

Substituting in formula **183,**

$$h = \frac{f l v_m^2}{5.36\, d} + .0233\, v_m^2 =$$

$$\frac{.02391 \times 2,000 \times (3.3)^2}{5.36 \times 2.5} + .0233 \times (3.3)^2 = 39.12 \text{ ft.}   \text{Ans.}$$

(**1424**)   (*b*)   80,000 gal. per hr. $= \dfrac{80,000}{60} = 1,333\frac{1}{3}$ gal.
per min.

To obtain the actual horsepower, apply formula **192.**

$H = .00038\, G\, h = .00038 \times 1,333\frac{1}{3} \times 420 = 212.8$ H. P.   Ans.

(*a*) The theoretical horsepower $= \frac{2}{3} \times 212.8 = 141.87$ H. P.
Ans.

(**1425**)   Applying formula **187** and substituting,

$$D = \frac{835.5\, G\, h}{W} = \frac{835.5 \times 30,000 \times 290}{600} = 12,114,750 \text{ ft.-lb.}$$
Ans.

(**1426**)   We first calculate the value of f from for-
mula **182** and the table.

$$v_m = 2.315 \sqrt{\frac{h\, d}{f\, l}} = 2.315 \sqrt{\frac{220 \times 6}{.025 \times 6,500}} = 6.597 \text{ ft. per sec.}$$

From the table, $f = .0214$ for $v_m = 6$ and .0205 for $v_m = 8.$

$\dfrac{.0214 - .0205}{2} = .00045 =$ decrease   for   an   increase   of

velocity of 1 ft. per sec. .00045 × .597 ft. per sec. = 000268, total decrease. .0214 − .000268 = .02113 = f.
Substituting in formula **182**,

$$v_{\blacksquare} = 2.315 \sqrt{\frac{220 \times 6}{.02113 \times 6,500}} = 7.17 \text{ ft. per sec.} \quad \textbf{Ans.}$$

(**1427**) (*a*) See Art. **2290**.
Head = 45 × 2.304 = 103.68 ft.   Ans.

(*b*) 2.304 × 86 = 198.144 ft.   Ans.

(*c*) 2.304 × 108 = 248.832 ft.   Ans.

(**1428**) (*b*) Applying formula **191** and substituting,
$G = .03264d^2 S = .03264 \times 15^2 \times 100 = 734.4$ gal. per min. **Ans.**

(*a*) To calculate the diameter of the steam-cylinder, we first obtain the horsepower from formula **192**, then substitute in formula **194**.

$$H = .00038\, G\, h = .00038 \times 734.4 \times 310 = 86 \text{ H. P.}$$

$$D = 205\sqrt{\frac{H}{PS}} = 205\sqrt{\frac{86}{50 \times 100}} = 27 \text{ in.} \quad \textbf{Ans.}$$

(**1429**) Applying formula **167** and substituting,
$v = \sqrt{2gh} = \sqrt{2 \times 32.16 \times 13.7} = 29.685$ ft. per sec. **Ans.**

(**1430**) According to Pascal's law, the pressure per square inch on each piston is the same. In order that the weights shall balance, they must be proportional to the area of the piston. Hence, we have the proportion

$$5 : 73 :: 22 : x.$$

$$x = \frac{22 \times 73}{5}, \text{ or } 321.2 \text{ lb.} \quad \textbf{Ans.}$$

(**1431**) As the lips are applied to the tube and the breath drawn in, the air in the tube above the surface of the water is drawn into the mouth, and a partial vacuum in the tube is the result of the operation. Now, as there is very little pressure on the water in the tube, and as the water outside

the tube is exposed to the pressure of the atmosphere, 14.7 lb. per sq. in., the water must be forced up the tube by the greater pressure of the atmosphere. The action known as suction is, therefore, only a manifestation of atmospheric pressure.

(1432) (*a*) Applying formula **191** and substituting,
$G = .03264\,d^2 S = .03264 \times 11^2 \times 100 = 394.944$ gal. per min.
   $394.944 \times 60 = 23,696.64$ gal. per hour.   Ans.

(*c*) Use formula **192.**
$H = .00038\,G\,h = .00038 \times 394.944 \times 300 = 45.024$ H. P. Ans.

(*b*) Applying formula **194,**

$$D = 205 \sqrt{\frac{H}{PS}} = 205 \sqrt{\frac{45.024}{50 \times 100}} = 19\tfrac{1}{2} \text{ inches.   Ans.}$$

(1433) Because the water helps to fill up the pores in the flat surface and the glass, thus creating a partial vacuum between the surfaces.

(1434) First finding the value of f by formula **182** and Table 45, we have

$$v_m = 2.315 \sqrt{\frac{h\,d}{f\,l}} = 2.315 \sqrt{\frac{15 \times 3.5}{.025 \times 88}} =$$
   11.3 ft. per sec., nearly.

From the table, $f = .0205$ for $v_m = 8$ and $.0193$ for $v_m = 12$.
$\dfrac{.0205 - .0193}{4} = .0003$, decrease for an increase of 1 ft. per sec.

   $.0003 \times 3.3 = .00099$, total decrease.
   $f = .0205 - .00099 = .01951.$

To obtain the discharge in gal. per sec., substitute this value of f in formula **186.**

$$Q = .09445\,d^2 \sqrt{\frac{h\,d}{f\,l + .125\,d}} =$$

$$.09445 \times (3.5)^2 \sqrt{\frac{15 \times 3.5}{.01951 \times 88 + .125 \times 3.5}} = 5.711 \text{ gal. per sec}$$

   $5.711 \times 60 = 342.66$ gal. per min.   Ans.

(**1435**) (*a*) To obtain the diameter of the steam-cylinder, we calculate the horsepower from formula **192,** then substitute in formula **194.**

$$H = .00038\, G\, h = .00038 \times 300 \times 225 = 25.65 \text{ H. P.}$$

$$D = 205 \sqrt{\frac{H}{PS}} = 205 \sqrt{\frac{25.65}{110 \times 50}} = 14 \text{ in.} \textbf{Ans.}$$

(*b*) Applying formula **190** and substituting,

$$d = 5.535 \sqrt{\frac{G}{S}} = 5.535 \sqrt{\frac{300}{110}} = 9\tfrac{1}{8} \text{ in.} \text{Ans.}$$

(*c*) Assume the number of strokes per minute to be 110; then, stroke $= \tfrac{110}{110} = 1$ ft. $= 12$ in.   Ans.

(*d*) Use formula **195.**

$$d_1 = .35 \sqrt{G} = .35 \sqrt{300} = 6 \text{ in.} \textbf{Ans.}$$

(*e*) Apply formula **196.**

$$d_2 = .25 \sqrt{G} = .25 \sqrt{300} = 4\tfrac{1}{3} \text{ in.} \textbf{Ans.}$$

(**1436**) (*a*) To obtain the theoretical discharge, apply formula **167.** $v = \sqrt{2\,g\,h} = \sqrt{2 \times 32.16 \times 15.75} = 31.83$ ft. per sec., since 15 ft. 9 in. $= 15.75$ ft.

31.83 $\times$ 60 $= 1,909.8$ ft. per min.

Area of orifice $= 11.2$ sq. in. $= \dfrac{11.2}{144}$ sq. **ft.**

To calculate the theoretical quantity in cu. ft. **per min.,** substitute in formula $Q = A\,v$.

$$Q = \frac{11.2}{144} \times 1,909.8 = 148.54 \text{ cu. ft. per min.} \textbf{Ans.}$$

(*b*) Applying formula **174** and substituting,

$$Q_a = .615 \times \frac{11.2}{144} \sqrt{2 \times 32.16 \times 15.75} = 1.5224 \text{ cu. ft. per sec.}$$

$$1.5224 \times 60 = 91.344 \text{ cu. ft. per min.} \textbf{Ans.}$$

(**1437**) Because if any air be left in the siphon, it will exert a pressure on the water in the arms of the siphon that will exactly balance the atmospheric pressure on the surface

of the water outside, which tends to force the water up the arms.

Therefore, the water in each arm is in equilibrium, and no motion can take place. As soon, however, as the air is expelled, either by filling the siphon with water or by pumping the air out, the water is no longer in equilibrium, and will begin to flow.

(1438)   Piston speed per minute = 9 × 5 = 45 ft.

(*a*) Applying formula **191**,

$G = .03264 d^2 S = .03264 \times 19^2 \times 45 = 530.24$ gal. per min.   Ans.

(*b*) 530.24 × 60 = 31,814.4 gal. per hr.   Ans.

(1439)   Applying formula **187** and substituting,

$$D = \frac{835.5 G h}{W} = \frac{835.5 \times 80,000 \times 340}{400} = 56,814,000 \text{ ft.-lb.} \quad \text{Ans.}$$

(1440)   See Art. **2290.**

(*a*) 2.304 × 80   = 184.32 ft.   Ans.

(*b*) 2.304 × 30.5 = 70.272 ft.   Ans.

(*c*) 2.304 × 108   = 248.832 ft.   Ans.

(*d*) 2.304 × 215   = 495.36 ft.   Ans.

(1441)   Applying formula **191** and substituting,

$G = .03264 d^2 S = .03264 \times 14^2 \times 100 = 639.744$ gal. per min.

639.744 × 60 = 38,384.64 gal. per hr., the delivery for one side.

Total delivery = 38,384.64 × 2 = 76,769.28 gal. per hr. Ans.

(1442) $f = .0205$ for $v_m = 8$.
Substituting in formula **183**,

$$h = \frac{f l v_m^2}{5.36 d} + .0233 v_m^2 = \frac{.0205 \times 5,000 \times 8^2}{5.36 \times 4} + .0233 \times 8^2 =$$
$$\frac{6,560}{21.44} + 1.49 = 307.46 \text{ ft.} \quad \text{Ans.}$$

(1443) Since the area of the orifice is greater than $\frac{1}{10}$ of the area of the cross-section of the vessel, we use formula **171.**

$$v = \sqrt{\frac{2gh}{1-\frac{a^2}{A^2}}} \cdot \sqrt{\frac{2 \times 32.16 \times 12}{1-\frac{(11 \times 11)^2}{(36^2 \times .7854)^2}}} = \sqrt{\frac{771.84}{1-.0141}} =$$

27.979 ft. per sec.   Ans.

(1444) Applying formula **187** and substituting,

$$D = \frac{835.5\,Gh}{W} = \frac{835.5 \times 4,000,000 \times 125}{7,460} = 55,998,660 \text{ ft.-lb.}$$
$$\text{Ans.}$$

(1445) Force available to accelerate the moving mass $=$ $20 - (12+3) = 5$ tons $= F$.   Weight to be accelerated $= 20 +$ $12 = 32$ tons $= W$.   By formula **188,** acceleration $= f = \frac{gF}{W} =$ $\frac{32.16 \times 5}{32} = 5.025$ ft. per sec.

By formula **189,** $t = \sqrt{\frac{2s}{f}} = \sqrt{\frac{2 \times 10}{5.025}} = 1.995$ sec. $=$ time occupied in passing over 10 feet.   This is at the rate of $\frac{10}{1.995} \times 60 = 300$ ft. per min.

Since the speed must not exceed 200 ft. per minute, the pit-work must be counterbalanced.   Suppose a counterweight of 2 tons be tried, assuming that the frictional resistances are not increased.

Then,   force $= F = 20 - (12 + 3 + 2) = 3$ tons.

Weight $= 20 + 12 + 2 = 34$ tons.

$$f = \frac{gF}{W} = \frac{32.16 \times 3}{34} = 2.84 \text{ ft. per sec.}$$

$$t = \sqrt{\frac{2s}{f}} = \sqrt{\frac{2 \times 10}{2.84}} = 2.65 \text{ sec., nearly.}$$

$$\frac{10}{2.65} \times 60 = 226.42 \text{ ft. per sec.}$$

This speed is also too great, so we will try a counterbalance of $2\frac{1}{2}$ tons.

$$\text{Force} = F = 20 - (12 + 3 + 2\frac{1}{2}) = 2\frac{1}{2} \text{ tons.}$$

$$\text{Weight} = W = 20 + 12 + 2\frac{1}{2} = 34\frac{1}{2} \text{ tons.}$$

$$f = \frac{gF}{W} = \frac{32.16 \times 2\frac{1}{2}}{34\frac{1}{2}} = 2.33 \text{ ft. per sec.}$$

$$t = \sqrt{\frac{2s}{f}}\sqrt{\frac{2 \times 10}{2.33}} = 2.93 \text{ sec., nearly.}$$

$$\frac{10}{2.93} \times 60 = 204.78 \text{ ft. per min.}$$

This is near enough for practical purposes, but if a counterweight of 2.6 tons be tried, it will reduce the acceleration so that the speed of the pit-work is almost exactly 200 ft. per min.

(1446) By the difference of cylinder volumes. The steam is admitted into the high-pressure cylinder and exhausted into the low-pressure cylinder.

(1447) See Art. **2269.**

(1448) Apply formulas **195** and **196.**

(*a*) $d_1 = .35 \sqrt{G} = .35 \sqrt{\dfrac{70,000}{60}} = 12$ in.   Ans.

(*b*) $d_2 = .25 \sqrt{G} = .25 \sqrt{\dfrac{70,000}{60}} = 8\frac{1}{2}$ in.   Ans.

(1449) 100,000 gal. per hr. $= \dfrac{100,000}{60}$ gal. per min.
Applying formula **192,**
$$H = .00038\, G h = .00038 \times \frac{100,000}{60} \times 480 = 304 \text{ H. P.}\quad \text{Ans.}$$

(1450) Apply formula **184.**
$$Q = .0408\, d^2 v_m = .0408 \times 7^2 \times .721 = 14.414232 \text{ gal. per sec.}$$
$$14.414232 \times 60 \times 60 = 51,891.24 \text{ gal. per hr.}\quad \text{Ans.}$$

(1451) See Art. **2260,**

(1452) See Art. **2271.**

(1453) See Arts. **2225, 2226, 2259, 2271,** and **2280.**

(1454) 200 ft. per min. ; 400 ft. per min. ; 100 ft. per min.

(1455) Applying formula **191** and substituting,

$G = .03264\, d^2\, S = .03264 \times 15^2 \times 95 =$ number of gal. per min.

$.03264 \times 15^2 \times 95 \times 60 = 41,860.8$ gal. per hr. **Ans.**

(1456) See Arts. **2250** to **2254.**

(1457) Applying formula **172** and substituting,

$$v = .98 \sqrt{2gh} = .98 \sqrt{2 \times 32.16 \times 69.12} = 65.34 \text{ ft. per sec.}$$
                                                                    Ans.

(1458) (*a*) Area of piston $= (\tfrac{7}{8})^2 \times .7854 = .6013$ sq. in.

Pressure per sq. in. exerted by piston $= \dfrac{50}{.6013} = 83.15$ lb.

A column of water 1 foot high and of 1 sq. in. cross-section weighs .434 pound, and therefore exerts a pressure of .434 pound per sq. in. The height of a column of water to exert a pressure of 83.15 lb. per sq. in. must be $\dfrac{83.15}{.434} = 191.6$ feet. Consequently, the water will rise 191.6 feet.

The diameter of the hole in the squirt-gun has nothing to do with the height of the water, since the pressure per square inch will remain the same, no matter what may be the diameter.

(*b*) Using formula **170,**

$$R = \sqrt{4hy} = \sqrt{4 \times 10 \times 191.6} = 87.54 \text{ ft.} \textbf{Ans.}$$

MINE HAULAGE.

(1459) See Art. **2298.**

(1460) See Art. **2299.**

(1461) See Art. **2300.**

(1462) See Art. **2301.**

(1463) See Art. **2303.**

(1464) See Art. **2304.**

(1465) See Art. **2305.**

(1466) See Arts. **2305** and **2311.**

(1467) See Art. **2306.**

(1468) See Art. **2307.**

(1469) See Art. **2308.**

(1470) See Art. **2309.**

(1471) See Art. **2310.**

(1472) See Art. **2311.**

(1473) See Art. **2312.**

(1474) See Art. **2312.**

(1475) See Art. **2312.**

(1476) See Art. **2313.**

(1477) See Art. **2314.**

(1478) As the hold or grip increases directly as the square of the number of coils, the proportion of grip the latter will have compared with the former is as $2^2 : 4^2$, or

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as 4 : 16; that is, the rope turned four times around the drum will have 4 times the grip or hold that the rope coiled twice around the drum has. See Art. **2315.**

(**1479**) It would equal two complete coils on one wheel, and the grip or haulage power would be four times that of a single coil around one wheel. See Art. **2315.**

(**1480**) See Arts. **2317** and **2318.**

(**1481**) See Art. **2319.**

(**1482**) See Art. **2320.**

(**1483**) See Art. **2321.**

(**1484**) See Art. **2322.**

(**1485**) 50 lb. See Art. **2324.**

(**1486**) See Art. **2324.**

(**1487**) See Art. **2325.**

(**1488**) Applying formula **197,** we have

$$.12 \times 3,500 + \frac{3,500}{40} = 507.5 \text{ lb. Ans.}$$

(**1489**) See Art. **2327.** By adding the weight of the empty car to that of the loaded car and dividing the sum by the coefficient of friction, or 40.

(**1490**) Applying formula **197,** we have $F_s = .10 \times 4,200 + \frac{4,200}{40} = 525$ lb., the force required to move the rope. Applying formula **199,** we have $F = .10 \times (4,000 - 1,800) - \frac{4,000 + 1,800}{40} = 75$ lb., the available gravity force due to one pair of cars. Therefore, the number of cars that must run in a train is $525 \div 75 = 7$ cars. Ans.

(**1491**) See Art. **2328.**

(**1492**) The rope weighs $250 \times 1.5 = 375$ lb. Applying formula **197,** we have $F_s = .25 \times 375 + \frac{375}{40} = 103.125$ lb.,

the power necessary to raise the rope. Applying formula **199,** and assuming that the loaded car is at the bottom of the jig and the balance car at the bottom of the plane, we have $F = .25 (4,000 - 2,900) - \dfrac{4,000 + 2,900}{40} = 102.5$ lb., the available gravity force at the descent of the full car. Now, as it requires 103.125 lb. to move the rope, and there is only 102.5 of gravity force available, it is plain that this jig will not operate.

(1493) See Art. **2328.**

(1494) See Art. **2329.**

(1495) See Art. **2329.**

$$\dfrac{(2 \times 1,800) + 1,200}{40} + .25 (2 \times 1,800 + 1,200) = 1,320 \text{ lb.}$$
<div align="right">**Ans.**</div>

(1496) See Arts. **2330** and **2331.**

(1497) See Arts. **2332** and **2333.**

(1498) See Art. **2337.**

(1499) See Art. **2339.**

(1500) See Art. **2340.**

(1501) See Art. **2342.**

(1502) See Art. **2343.**

(1503) See Art. **2343.**

(1504) See Art. **2344.**

(1505) See Art. **2345.**

(1506) See Art. **2346.**

(1507) See Art. **2347.**

(1508) Applying formula **200,**

$$T = \dfrac{(18 \times 4,000) + (5,000 \times .88)}{40} + .05 (18 \times 4,000 + 5,000 \times .88) =$$
5,730 lb. Ans.

(1509) The velocity of the train is $\dfrac{10 \times 5,280}{60} = 880$ ft. per min. $880 \times 5,730 = 5,042,400 =$ foot-pounds of work per minute required of the engine. $5,042,400 \div 33,000 = 152.8$ H. P.   Ans.

(1510) See Art. **2349.**

(1511) See Art. **2350.**

(1512) See Art. **2351.**

(1513) The weight of the rope $= 3,000 \times 2 \times .88 = 5,280$ lb. The weight of 25 empty cars weighing 1,500 lb. each $= 1,500 \times 25 = 37,500$ lb.; therefore, the resistance due to friction $= \dfrac{5,280 + 37,500}{40} = 1,069.5$ lb. The resistance due to gravity $= 37,500 \times .05 = 1,875$ lb. Then, $1,069.5 + 1,875 = 2,944.5$ lb., the tension on the rope. 12 miles per hour $= \dfrac{5,280 \times 12}{60} = 1,056$ ft. per minute.

$$\frac{1,056 \times 2,944.5}{33,000} = 94.2 \text{ H. P.}\quad \text{Ans.}$$

Or, by formula **201,** the tension can be found as follows:

$$T = \frac{37,500 + 5,280}{40} + .05 \times 37,500 = 2,944.5 \text{ lb.}$$

The H. P. can be found by formula **202,** as follows:

$$H = \frac{2,944.5 \times 1,056}{33,000} = 94.2 \text{ H. P.}\quad \text{Ans.}$$

(1514) See Art. **2354.**

(1515) See Art. **2355.**

(1516) See Art. **2356.**   .

(1517) See Arts. **2357** and **2358.**

(1518) See Art. **2358.**

(1519) See Art. **2359.**

(1520) See Art. **2360.**

(1521) See Art. **2365.**

(1522) As the roads are level, there is no tension due to grade, and formula **201** becomes simply $T = \dfrac{W + w'}{40} = \dfrac{90,000 + 9,100}{40} = 2,477.5$ lb., the tension in the main rope.

Ans.

To find the tension in the tail-rope, the weight of the train of empty cars is found.

Then, $T_1 = \dfrac{30,000 + 9,100}{40} = 977.5$ lb.   Ans.

As the conditions of the problem require the maximum tension on the rope, we take that on the main rope, or 2,477.5 lb., and applying formula **202**,

$$H = \frac{2,477.5 \times \dfrac{(10 \times 5,280)}{60}}{33,000} = \frac{2,180,200}{33,000} = 66.1 \text{ H. P.} \text{ Ans.}$$

(1523) Applying formula **201**, we have

$$T = \frac{90,000 + 12,480}{40} + .03 \times 90,000 = 5,262 \text{ lb.} \text{ Ans.}$$

(1524) Applying formula **203**, we have

$$P_1 = \frac{90\,(6,000 + 60)}{6,000} = 90.9 \text{ H. P.}$$

(1525) As the gravity force due to the pitch of the incline reduces the tension on the main rope, it must be treated negatively.   Then, formula **201** becomes

$$T = \frac{W + w}{40} - a \times W = \frac{100,000 + 7,040}{40} - .04 \times 100,000 =$$
$- 1,324$ lb., or the negative tension on the main rope.   Ans.

This means that there is not only no tension on the main rope, but an excess of gravity force equal to 1,324 lb.   The gravity force in the case of hauling the train of empty cars is positive, and can be found by use of formula **201.**

$$T = \frac{40,000 + 7,040}{40} + .04 \times 40,000 = 2,776 \text{ lb.,}$$

the tension of the tail-rope.   Ans.

No horsepower is exerted on the main rope, because, as shown previously, the tension is negative. By using formula **202**, the horsepower exerted over the tail-rope is

$$H = \frac{2{,}776 \times \dfrac{(5{,}280 \times 11)}{60}}{33{,}000} = 81.4 \text{ H. P.} \quad \text{Ans.}$$

(1526) By the use of formula **204**, we have $T = \dfrac{5{,}000 \times 20 + 4{,}000 (3.65 + .6)}{40} - .04 \times (100{,}000 - 12{,}200) =$ $(2{,}925 - 3{,}512) = -587$ lb., the negative tension. For the tension in the tail-rope, formula **205** is used. $T_1 = \dfrac{40{,}000 + 4{,}000 (3.65 + 6)}{40} + .04 (40{,}000 - 12{,}200) = 1{,}425 +$ $1{,}112 = 2{,}537$ lb., the tension on the tail-rope. Now, as the tension on the main rope is negative, there is no power applied to it; on the tail-rope, however, in which there is a tension of 2,537 lb. with the trains running 11 miles per hour, we have H. P. $= \dfrac{2{,}537 \times \dfrac{(5{,}280 \times 11)}{60}}{33{,}000} = 74.4 \text{ H.P.} \quad \text{Ans.}$

(1527) See Art. **2372.**

(1528) See Art. **2372.**

(1529) See Art. **2373.**

(1530) $6{,}000 + 4{,}800 + 2{,}500 + 7{,}000 + 3{,}000 = 23{,}300;$ $\dfrac{23{,}300}{5} = 4{,}660$, the mean length. $\dfrac{5{,}280 \times 12 \times 10}{4{,}660 \times 3} = 45.3,$ practically 46 trains. Ans.

(1531) $\dfrac{2{,}500}{46 \times 2.5} = 21.7$, say 22 cars. Ans.

(1532) Allowing $\frac{1}{3}$ of the time for stoppage, the rope travels for $\frac{2}{3}$ of $10 = 6\frac{2}{3}$ hours, and hauls coal for but $\frac{1}{2}$ this time, or $3\frac{1}{3}$ hours. Hence, the distance the rope travels while hauling coal is $5{,}280 \times 11 \times 3\frac{1}{3} = 193{,}600$ feet, and since the mean length of the haulage roads, which is found by dividing their total length by 4, is

$$\frac{4,250 + 3,012 + 756 + 514}{4} = 2,133 \text{ feet,}$$

the number of loaded trains is

$$\frac{193,600}{2,133} = 90.76, \text{ or } 91. \quad \textbf{Ans.}$$

The number of cars in each train is

$$\frac{2,500}{91 \times 2.5} = 11. \quad \textbf{Ans.}$$

The weight of the rope is equal to its weight per **foot** multiplied by twice the maximum haul, or $4,250 \times 2 \times 1.5 =$ 12,750 pounds, and the weight of a loaded car is $2,000 +$ $2.5 \times 2,000 = 7,000$ pounds.

Substituting in formula **201,**

$$T = \frac{7,000 \times 11 + 12,750}{40} + .03 \times 77,000 = 4,553.75 \text{ pounds.}$$

The speed of the train is equal to

$$\frac{5,280 \times 11}{60} = 968 \text{ feet per minute.}$$

Applying formula **202,**

$$H = \frac{4,553.75 \times 968}{33,000} = 133.6 \text{ horsepower, nearly.} \quad \textbf{Ans.}$$

(1533)   See Art. **2374.**

(1534)   See Art. **2375.**

(1535)   See Art. **2375.**

(1536)   See Art. **2376.**

(1537)   See Art. **2377.**

(1538)   See Art. **2377.**

(1539)   See Art. **2377.**

(1540)   See Art. **2378.**

(1541)   See Art. **2379.**

(1542) See Art. **2379.**

(1543) See Art. **2380.**

(1544) See Arts. **2384** and **2385.**

(1545) See Arts. **2384** and **2386.**

(1546) See Art. **2382.**

(1547) See Art. **2382.**

(1548) See Art. **2387.**

(1549) See Art. **2387.**

(1550) See Arts. **2387** and **2388.**

(1551) See Art. **2388.**

(1552) See Arts. **2390** to **2392.**

(1553) See Art. **2395.**

(1554) See Art. **2395.**

(1555) See Arts. **2397** and **2398.**

(1556) See Art. **2401.**

(1557) By formula **207,** the number of cars on the rope is $\dfrac{2,500 \times 5,230}{2 \times 5,280 \times 8 \times 1.5} = 103.18.$ say 103.   Ans.

And, by formula **208,** the distance the cars are apart is $\dfrac{5,230}{103.18} = 50.68$ ft.   Ans.

(1558) See Art. **2406.**

(1559) See Art. **2407.**

(1560) See Arts. **2408** and **2409.**

(1561) See Art. **2411.**

(1562) By formula **207,** the number of loaded cars on the rope at one time is $\dfrac{976 \times 4,720}{2.5 \times 5,280 \times 10 \times 1.25} = 27.919.$ The weight of the loaded cars on one side of the rope will then be $4,000 \times 27.919 = 111,676$ pounds. Taking the weight of an empty car at 1,200 pounds, the weight of the empty cars on

the rope will be $1,200 \times 27.919 = 33,502.8$. The weight of the rope is $4,720 \times 2 \times 3 = 28,320$ pounds. Then, substituting in formula **210,** $T = \dfrac{(111,676 + 33,502.8 + 28,320)}{40} = $ $4,337.47$ pounds, the tension on the rope. Ans.

A velocity of $2\frac{1}{2}$ miles an hour is equal to $\dfrac{2.5 \times 5,280}{60} = $ 220 feet per minute. Using formula **202,** the horsepower is
$$H = \frac{4,337.47 \times 220}{33,000} = 28.92 \text{ H. P.} \quad \text{Ans.}$$

(1563) As the two sides of the rope balance each other and the cars balance each other, only the weight of the coal is subject to the gravity of the grade. Substituting in formula **210,** we have

$$T = \frac{(111,676 + 33,502.8 + 28,320)}{40} + .025(111,676 - 33,502.8) =$$

$6,291.8$ pounds tension on the rope. Ans.

The velocity is $\dfrac{2.5 \times 5,280}{60} = 220$ feet per minute. The horsepower is, therefore, $\dfrac{6,291.8 \times 220}{33,000} = 41.9$ horsepower.
Ans.

(1564) Substituting in formula **210,** we have

$$T = \frac{(111,676 + 33,502.8 + 28,320)}{40} - .025(111,676 - 33,502.8) =$$

$2,383.14$ lb., the tension on the rope. Ans.

The horsepower required is $\dfrac{2,383.14 \times 220}{33,000} = 15.89$ horse power. Ans.

(1565) See Art. **2414.**

(1566) See Arts. **2414** and **2415.**

(1567) See Art. **2418.**

(1568) See Art. **2420.**

(1569)   See Art. **2422.**

(1570)   See Arts. **2428** to **2430.**

(1571)   See Art. **2431.**

(1572)   See Arts. **2432** and **2433.**

(1573)   See Art. **2435.**

(1574)   See Art. **2437.**

(1575)   See Art. **2439.**

(1576)   See Art. **2440.**

(1577)   See Art. **2443.**

(1578)   See Art. **2444.**

(1579)   See Art. **2446.**

(1580)   See Art. **2446.**

(1581)   See Art. **2449.**

(1582)   See Art. **2450.**

(1583)   See Art. **2450.**

(1584)   See Art. **2316.**

(1569) See Art. 2422.

(1570) See Arts. 2428 to 2430.

(1571) See Art. 2431.

(1572) See Arts. 2432 and 2433.

(1573) See Art. 2435.

(1574) See Art. 2437.

(1575) See Art. 2439.

(1576) See Art. 2440.

(1577) See Art. 2443.

(1578) See Art. 2444.

(1579) See Art. 2440.

(1580) See Art. 2446.

(1581) See Art. 2449.

(1582) See Art. 2450.

(1583) See Art. 2450.

(1584) See Art. 2316.

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